CHAPTER B4

STRESS ANALYSIS OF PIPING SYSTEMS

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Piping stress analysis is a discipline which is highly interrelated with piping layout (Chap. B3) and support design (Chap. B5). The layout of the piping system should be performed with the requirements of piping stress and pipe supports in mind (i.e., sufficient flexibility for thermal expansion; proper pipe routing so that simple and economical pipe supports can be constructed; and piping materials and section properties commensurate with the intended service, temperatures, pressures, and anticipated loadings). If necessary, layout solutions should be iterated until a satisfactory balance between stresses and layout efficiency is achieved. Once the piping layout is finalized, the piping support system must be determined. Possible support locations and types must be iterated until all stress requirements are satisfied and other piping allowables (e.g., nozzle loads, valve accelerations, and piping movements) are met. The piping supports are then designed (Chap. B5) based on the selected locations and types and the applied loads.

This chapter discusses several aspects of piping stress analysis. The discussion is heavily weighted to the stress analysis of piping systems in nuclear power plants, since this type of piping has the most stringent requirements. However, the discussion is also applicable to the piping systems in ships, aircraft, commercial buildings, equipment packages, refrigeration systems, fire protection piping, petroleum refineries, and so on. Each of these types of piping must meet the requirements of its applicable code.

FAILURE THEORIES, STRESS CATEGORIES, STRESS LIMITS, AND FATIGUE

Failure Theories

The failure theories most commonly used in describing the strength of piping systems are the *maximum principal stress theory* and the *maximum shear stress theory* (also known as the *Tresca criterion*).

The maximum principal stress theory forms the basis for piping systems governed by ASME B31 and Subsections NC and ND (Classes 2 and 3) of Section III of the ASME Boiler and Pressure Vessel Codes. This theory states that yielding in a piping component occurs when the magnitude of any of the three mutually perpendicular principal stresses exceeds the yield strength of the material.

The maximum shear stress theory is more accurate than the maximum principal stress theory for predicting both yielding and fatigue failure in ductile metals. This maximum shear stress theory forms the basis for piping of Subsection NB (Class 1) of ASME Section III.[1]

The maximum shear stress at a point τ_{max} is defined as one-half of the algebraic difference between the largest and the smallest of the three principal stresses σ_1, σ_2, and σ_3. If $\sigma_1 > \sigma_2 > \sigma_3$ (algebraically), then $\tau_{max} = (\sigma_1 - \sigma_3)/2$. The maximum shear stress theory states that failure of a piping component occurs when the maximum shear stress exceeds the shear stress at the yield point in a tensile test. In the tensile test, at yield, $\sigma_1 = S_y$ (yield stress), $\sigma_2 = \sigma_3 = 0$. So yielding in the component occurs when

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{S_y}{2} \tag{B4.1}$$

Equation (B4.1) has an unnecessary operation of dividing both sides by 2 before comparing them. For the sake of simplicity, a stress defined as $2\tau_{max}$ and equal to $\sigma_{max} - \sigma_{min}$ of the three principal stresses has been used for Class 1 piping. This stress is called the *equivalent intensity of combined stresses*, or *stress intensity*. Thus the stress intensity S is directly comparable to the tabulated yield stress values S_y from tensile tests with some factor of safety.

Stress Categories

There are various failure modes which could affect a piping system. The piping engineer can provide protection against some of these failure modes by performing stress analysis according to the piping codes. Protection against other failure modes is provided by methods other than stress analysis. For example, protection against brittle fracture is provided by material selection. The piping codes address the following failure modes: excessive plastic deformation, plastic instability or incremental collapse, and high-strain–low-cycle fatigue. Each of these modes of failure is caused by a different kind of stress and loading. It is necessary to place these stresses into different categories and set limits to them.

STRESS ANALYSIS OF PIPING SYSTEMS

The major stress categories *are primary, secondary, and peak*. The limits of these stresses are related to the various failure modes as follows:

1. The primary stress limits are intended to prevent plastic deformation and bursting.
2. The primary plus secondary stress limits are intended to prevent excessive plastic deformation leading to incremental collapse.
3. The peak stress limit is intended to prevent fatigue failure resulting from cyclic loadings.

Primary stresses which are developed by the imposed loading are necessary to satisfy the equilibrium between external and internal forces and moments of the piping system. Primary stresses are not self-limiting. Therefore, if a primary stress exceeds the yield strength of the material through the entire cross section of the piping, then failure can be prevented only by strain hardening in the material. Thermal stresses are never classified as primary stresses. They are placed in both the secondary and peak stress categories.

Secondary stresses are developed by the constraint of displacements of a structure. These displacements can be caused either by thermal expansion or by outwardly imposed restraint and anchor point movements. Under this loading condition, the piping system must satisfy an imposed strain pattern rather than be in equilibrium with imposed forces. Local yielding and minor distortions of the piping system tend to relieve these stresses. Therefore, secondary stresses are self-limiting. Unlike the loading condition of secondary stresses which cause distortion, peak stresses cause no significant distortion. Peak stresses are the highest stresses in the region under consideration and are responsible for causing fatigue failure. Common types of peak stresses are stress concentrations at a discontinuity and thermal gradients through a pipe wall.

Primary stresses may be further divided into general primary membrane stress, local primary membrane stress, and primary bending stress. The reason for this division is that, as will be discussed in the following paragraph, the limit of a primary bending stress can be higher than the limit of a primary membrane stress.

Basic Stress Intensity Limits

The basic stress intensity limits for the stress categories just described are determined by the application of limit design theory together with suitable safety factors.

The piping is assumed to be elastic and perfectly plastic with no strain hardening. When this pipe is in tension, an applied load producing a general primary membrane stress equal to the yield stress of the material S_y results in piping failure. Failure of piping under bending requires that the entire cross section be at this yield stress. This will not occur until the load is increased above the yield moment of the pipe multiplied by a factor known as the *shape factor* of the cross section. The shape factor for a simple rectangular section in bending is 1.5.

When a pipe is under a combination of bending and axial tension, the limit load depends on the ratio between bending and tension. In Fig. B4.1, the limit stress at the outer fiber of a rectangular bar under combined bending and tension is plotted against the average tensile stress across the section. When the average tensile stress P_m is zero, the failure bending stress is 1.5 S_y. When P_m alone is applied (no bending stress P_b), failure stress is yield stress S_y.

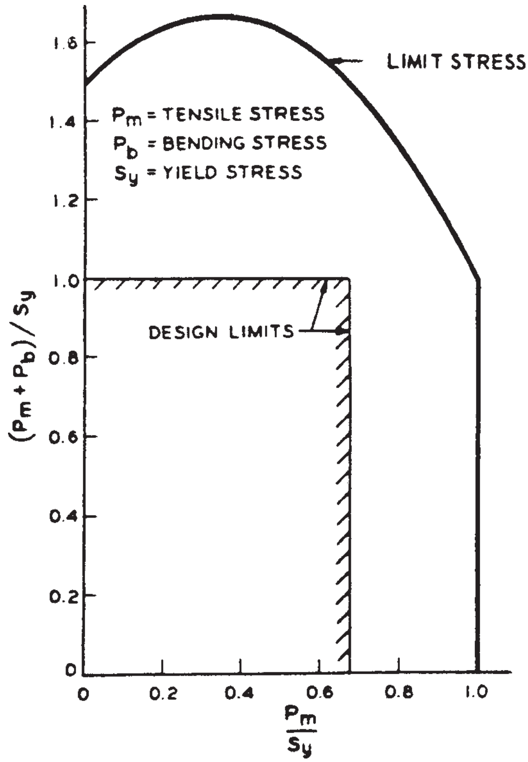


FIGURE B4.1 Limit stress for combined tension and bending (rectangular section). (*ASME, "Criteria."*[1] *Courtesy of ASME.*)

It also can be seen in Fig. B4.1 that a design limit of $^2/_3\ S_y$ for general primary membrane stress P_m and a design limit of S_y for primary membrane-plus-bending stress $P_m + P_b$ provide adequate safety to prevent yielding failure.

For secondary stresses, the allowable stresses are given in terms of a calculated elastic stress range. This stress range can be as high as twice the yield stress. The reason for this high allowable stress is that a repetitively applied load which initially stresses the pipe into plastic yielding will, after a few cycles, "shake it down" to elastic action.

This statement can be understood by considering a pipe which is strained in tension to a point e_1 somewhat beyond its yield strain, as shown in Fig. B4.2. The calculated elastic stress at this point would be equal to the product of the modulus of elasticity E and the strain ε_1, or $S_1 = E\varepsilon_1$. The path $OABC$ is considered as cycling the strain from 0 to ε_1 (loading) and back to 0 (unloading). When the pipe is returned to its original position O, it will retain a residual compressive stress of magnitude $S_1 - S_y$. On each subsequent loading cycle, this residual compression must be overcome before the pipe can go into tension; thus the elastic range has been extended by the value $S_1 - S_y$.

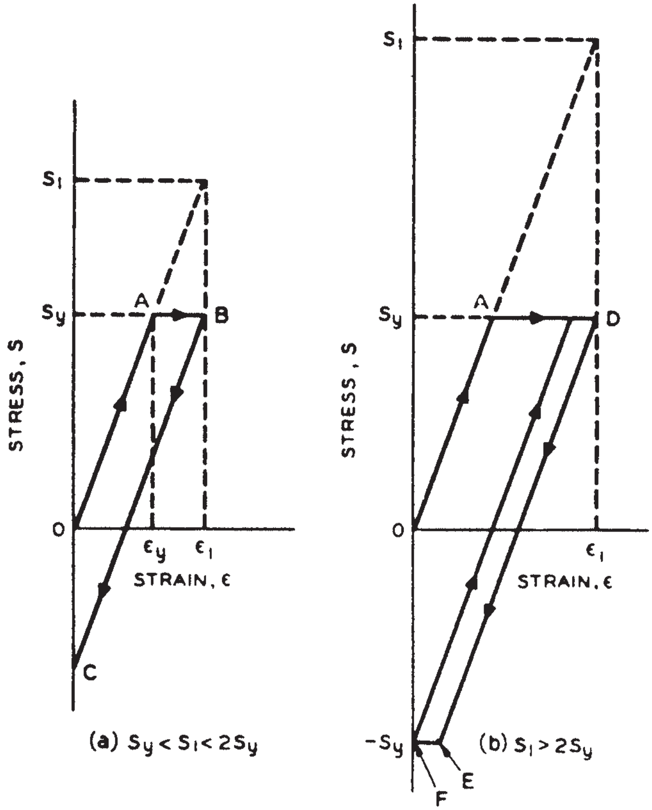


FIGURE B4.2 Strain history beyond yield. (*ASME, "Criteria."*[1] *Courtesy of ASME.*)

Therefore, the allowable secondary stress range can be as high as $2S_y$ when $S_1 = 2S_y$. When $S_1 > 2S_y$, the pipe yields in compression and all subsequent cycles generate plastic strain *EF*. For this reason $2S_y$ is the limiting secondary stress which will shake down to purely elastic action.

Fatigue

As mentioned previously, peak stresses are the highest stresses in a local region and are the source of fatigue failure. The fatigue process may be divided into three stages: *crack initiation* resulting from the continued cycling of high stress concentrations, *crack propagation* to critical size, and *unstable rupture* of the remaining section.

Fatigue has long been a major consideration in the design of rotating machinery, where the number of loading cycles is in the millions and can be considered infinite for all practical purposes. This type of fatigue is called *high-cycle fatigue*. High-cycle fatigue

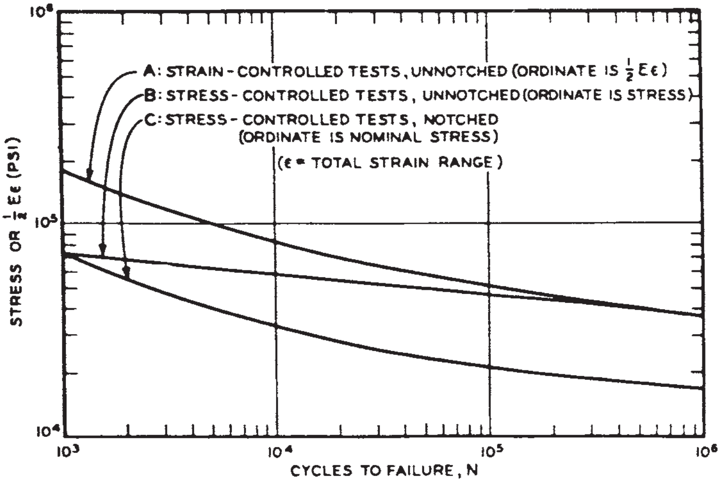


FIGURE B4.3 Typical relationship among stress, strain, and cycles to failure. (*ASME, "Criteria."*[1] *Courtesy of ASME.*)

involves little or no plastic action. Therefore, it is stress-governed. For every material, a fatigue curve, also called the *S–N curve*, can be generated by experimental test[2] which correlates applied stress with the number of cycles to failure, as shown in Fig. B4.3. For high-cycle fatigue, the analysis is to determine the endurance limit, which is the stress level that can be applied an infinite number of times without failure.

In piping design, the loading cycles applied seldom exceed 10^5 and are frequently only a few thousand. This type of fatigue is called *low-cycle fatigue*. For low-cycle fatigue, data resulting from experimental tests with stress as the controlled variable are considerably scattered. These undesirable test results are attributable to the fact that in the low-cycle region the applied stress exceeds the yield strength of the material, thereby causing plastic instability in the test specimen.

However, when strain is used as the controlled variable, the test results in this low-cycle region are consistently reliable and reproducible.

As a matter of convenience, in preparing fatigue curves, the strains in the tests are multiplied by one-half the elastic modulus to give a pseudostress amplitude. This pseudostress is directly comparable to stresses calculated on the assumption of elastic behavior of piping. In piping stress analysis, a stress called the *alternating stress* S_{alt} is defined as one-half of the calculated peak stress. By ensuring that the number of load cycles N associated with a specific alternating stress is less than the number allowed in the *S–N* curve, fatigue failure can be prevented. However, practical service conditions often subject a piping system to alternating stresses of different magnitudes. These changes in magnitude make the direct use of the fatigue curves inapplicable since the curves are based on constant-stress amplitude. Therefore, to make fatigue curves applicable for piping, it is necessary to take some other approach.

One method of appraising the fatigue failure in piping is to assume that the cumulative damage from fatigue will occur when the cumulative usage factor U equals unity, i.e.,

$$U = \Sigma U_i = \Sigma \frac{n_i}{N_i} = 1 \tag{B4.2}$$

where U_i = usage factor at stress level i
n_i = number of cycles operating at stress level i
N_i = number of cycles to failure at stress level i per fatigue curve

CLASSIFICATION OF LOADS, SERVICE LIMITS, AND CODE REQUIREMENTS

Classification of Loads

Primary loads can be divided into two categories based on the duration of loading. The first category is *sustained loads*. These loads are expected to be present throughout normal plant operation. Typical sustained loads are pressure and weight loads during normal operating conditions. The second category is *occasional loads*. These loads are present at infrequent intervals during plant operation. Examples of occasional loads are earthquake, wind, and fluid transients such as water hammer and relief valve discharge.

In addition to primary loads, there are *expansion loads*. Expansion loads are those loads due to displacements of piping. Examples are thermal expansion, seismic anchor movements, thermal anchor movements, and building settlement.

Service Limits

Service levels and their limits are defined for nuclear power plant safety-related piping by the ASME Boiler and Pressure Vessel Code, Section III.[3] They are described in the following list:

1. *Level A service limits*. The piping components or supports must satisfy these sets of limits in the performance of their specified service function. Examples of level A loadings are operating pressure and weight loadings.
2. *Level B service limits*. The piping component or support must withstand these loadings without damage requiring repair. Examples of level B loadings are fluid transients such as water hammer and relief valve discharge, and *operating-basis earthquake (OBE),* defined as the maximum likely earthquake postulated to occur during plant design life or one-half of the safe shutdown earthquake (see definition below), whichever is higher.
3. *Level C service limits*. The occurrence of stress up to these limits may necessitate the removal of the piping component from service for inspection or repair of damage. An example of level C loading is the combination of fluid transient loads occurring simultaneously with the operating-basis earthquake.
4. *Level D service limits*. These sets of limits permit gross general deformations with some consequent loss of dimensional stability and damage requiring repair, which may require removal of the piping component from service. An example of level D loading is the loading associated with a loss-of-coolant accident or a

safe-shutdown earthquake (SSE), which is defined as the maximum possible earthquake postulated to occur at the site of the plant at any time.

Code Requirements

There are various ASME and ANSI codes which govern the stress analysis of diferent kinds of pressure piping. These codes contain basic reference data, formulas, and equations necessary for piping design and stress analysis.

Each power plant is committed to a particular edition of a code for different types of piping. For example, the nuclear Class 1, 2, and 3 piping of a power plant may be committed to comply with the ASME Boiler and Pressure Vessel Code, Section III, 1974 edition, while the nonnuclear piping may be committed to ANSI B31.1 *Power Piping Code*, 1973 edition.

The following sections provide summaries of the ASME and ANSI codes.

ASME Boiler and Pressure Vessel Code, Section III, Subsection NB.[3] This subsection provides the code requirements of nuclear piping designated as Class 1. The loadings required to be considered for this subsection are the effects of pressure, weight (live and dead loads), thermal expansion and contraction, impact, earthquake, and vibrations. The stress limits which must be met are as follows:

1. *Primary stress intensity.* The primary stress intensity must meet the following requirement:

$$B_1\left(\frac{PD_o}{2t}\right) + B_2\left(\frac{D_o}{2I}\right)M_i \le kS_m \tag{B4.3}$$

where B_1, B_2 = primary stress indices for specific piping components under investigation
P = design pressure, psi
D_o = outside diameter of pipe, in
t = nominal wall thickness, in
M_i = resultant moment due to combination of design mechanical loads, in · lb
I = moment of inertia, in^4
kS_m = $1.5S_m$ for service level A; $1.8S_m$ for service level B but not greater than $1.5Sy$; $2.25S_m$ for service level C but not greater than $1.8S_y$; and $3.0S_m$ for service level D but not greater than 2.0Sy
S_m = allowable design stress intensity, psi
S_y = yield strength value taken at average fluid temperature under consideration, psi

2. *Primary plus secondary stress intensity range.* The following equations are used to evaluate a stress range as the piping system goes from one service load set (pressure, temperature, and moment) to any other service load set which follows in time. For each specified pair of load sets, the stress range S_n is calculated:

$$S_n = C_1\left(\frac{P_0D_o}{2t}\right) + C_2\left(\frac{D_o}{2I}\right)M_i + C_3E_{ab}|\alpha_aT_a - \alpha_bT_b| \tag{B4.4}$$

where C_1, C_2, C_3 = secondary stress indices for specific component under consideration

P_0 = range of service pressure, psi
M_i = resultant range of moment, in · lb
E_{ab} = average modulus of elasticity of two sides of a gross structural discontinuity or material discontinuity at room temperature , psi
α_a, α_b = coefficient of thermal expansion on side a or b of gross structural discontinuity or material discontinuity at room temperature , in/(in · °F)
T_a, T_b = range of average temperature on side a or b of gross structural discontinuity or material discontinuity, °F

And S_n has the following limit: $S_n \le 3S_m$.

If this requirement is not met for all pairs of load sets, then the piping component may still be qualified by using the simplified elastic-plastic discontinuity analysis described below; otherwise, the stress analyst may proceed to the fatigue analysis.

3. *Simplified elastic-plastic discontinuity analysis*. If $S_n > 3S_m$ for some pairs of load sets, a simplified elastic-plastic analysis may be performed if the thermal stress ratchet is not present. This analysis is required only for the specific load sets which exceeded $3S_m$. The following two equations must be satisfied:

$$S_e = C_2\left(\frac{D_o}{2I}\right)M_i^* \le 3S_m \qquad \text{(B4.5)}$$

$$C_1\left(\frac{P_0 D_o}{2t}\right) + C_2\left(\frac{D_o}{2I}\right)M_i + C_3 E_{ab}|\alpha_a T_a - \alpha_b T_b| \le 3S_m$$

where S_e = nominal value of expansion stress, psi
M_i^* = resultant range of moments due to thermal expansion and thermal anchor movements, in · lb
M_i = resultant range of moment excluding moments due to thermal expansion and thermal anchor movements, in · lb
C_3 = stress index for specific component under consideration

For later editions of the code, if $S_n > 3S_m$, the thermal stress ratchet must be evaluated and demonstrated to be satisfactory before a simplified elastic-plastic discontinuity analysis can be done. This ratchet is a function of the $|\Delta T_1|$ (see definition below) range only. The following requirement must be met:

$$|\Delta T_1| \text{ range } \le \frac{y' S_y C_4}{0.7E\alpha} \qquad \text{(B4.6)}$$

where

$$y' = \begin{array}{ll} 3.33 & \text{for } x = 0.3 \\ 2.00 & \text{for } x = 0.5 \\ 1.20 & \text{for } x = 0.7 \\ 0.80 & \text{for } x = 0.8 \end{array}$$

$x = PD_o/(2tS_y)$
S_y = yield strength value taken at average fluid temperature under consideration, psi
C_4 = 1.1 for ferritic material and 1.3 for austenitic material
$E\alpha$ = modulus of elasticity E times mean coefficient of thermal expansion α, both at room temperature, psi/°F
P = maximum pressure for set of conditions under consideration, psi

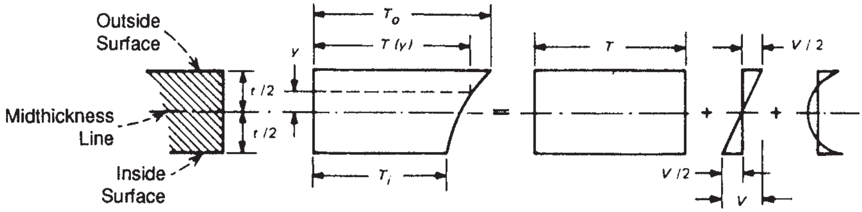


FIGURE B4.4 Decomposition of temperature distribution range. (*Figure NB-3653.2(b)-1, Section III, Division 1, ASME B & PV Code, 1989. Courtesy of ASME.*)

4. *Peak stress intensity range and fatigue analysis.* For each specified loading condition, peak stress is calculated as follows:

$$S_p = K_1 C_1 \left(\frac{P_0 D_o}{2t}\right) + K_2 C_2 \left(\frac{D_o}{2I}\right) M_i + \frac{K_3 E\alpha |\Delta T_1|}{2(1-\nu)} + K_3 C_3 E_{ab} |\alpha_a T_a - \alpha_b T_b| + \frac{E\alpha |\Delta T_2|}{1-\nu} \tag{B4.7}$$

where K_1, K_2, K_3 = local stress indices for specific component under consideration

ν = Poisson's ratio of material

$|\Delta T_1|$ = absolute value of range of temperature difference between temperature of outside surface and inside surface of pipe wall, assuming moment generating equivalent linear temperature distribution, °F (see Fig. B4.4)

$|\Delta T_2|$ = absolute value of range of that portion of nonlinear thermal gradient through wall thickness not included in ΔT_1, °F (see Fig. B4.4)

For each S_p, an alternating stress intensity S_{alt} is determined by

$$S_{alt} = \frac{K_e S_p}{2} \tag{B4.8}$$

where

$$K_e = \begin{cases} 1.0 & \text{for } S_n \le 3S_m \\ 1.0 + (1-n)[S_n/(3S_m) - 1]/[n(m-1)] & \text{for } 3S_m < S_n < 3mS_m \\ 1/n & \text{for } S_n \ge 3mS_m \end{cases}$$

m, n = material parameters given in Table B4.1

The alternating stress intensities are used to evaluate the cumulative effect of the stress cycles on the piping system. This evaluation is performed as follows:

a. The number of times each stress cycle of type 1, 2, 3, etc., is repeated during the life of the system shall be called n_1, n_2, n_3, and so on. Cycles shall be superimposed such that the maximum possible peak stress ranges are developed.

TABLE B4.1 Values of m, n, and T_{max} for Various Classes of Permitted Materials

Materials	m	n	T_{max}, °F
Carbon steel	3.0	0.2	700
Low-alloy steel	2.0	0.2	700
Martensitic stainless steel	2.0	0.2	700
Austenitic stainless steel	1.7	0.3	800
Nickel-chromium-iron	1.7	0.3	800
Nickel-copper	1.7	0.3	800

Source: Table NB-3228.5(b)-1, Section III, Division 1, ASME B & PV Code, 1998. (*Courtesy of ASME.*)
Note: (°F - 32)/1.8 = °C).

b. For each type of stress cycle, determine the alternating stress intensity S_{alt}.

c. For each value of S_{alt}, use the applicable design fatigue curve from the code to determine the maximum number of cycles permitted if this were the only cycle occurring. These numbers shall be designated N_1, N_2, N_3, and so on.

d. For each type of stress cycle, calculate the usage factor:

$$U_1 = \frac{n_1}{N_1} \qquad U_2 = \frac{n_2}{N_2} \qquad U_3 = \frac{n_3}{N_3} \qquad \cdots$$

e. The cumulative usage factor U is the sum of the individual usage factors:

$$U = U_1 + U_2 + U_3 + \cdots$$

ASME Boiler and Pressure Vessel Code, Section III, Subsections NC and ND.[3] These two subsections give the code requirements of nuclear piping designated as Class 2 and Class 3, respectively. The loadings required to be considered for Subsections NC and ND are the effects of pressure, weight, other sustained loads, thermal expansion and contraction, and occasional loads. The stress limits to be met are as follows:

1. *Stresses due to sustained loads*. The calculated stresses due to pressure, weight, and other sustained mechanical loads must meet the allowable $1.5S_h$, that is,

$$\frac{B_1 P D_o}{2t} + \frac{B_2 M_A}{Z} \leq 1.5 S_h \tag{B4.9}$$

where P = internal design pressure, psi
D_o = outside diameter of pipe, in
t = nominal wall thickness, in
Z = section modulus of pipe, in^3
M_A = resultant moment loading on cross section due to weight and other sustained loads, in · lb
S_h = basic material allowable stress at design temperature, psi

2. *Stresses due to occasional loads.* The calculated stress due to pressure, weight, other sustained loads, and occasional loads must meet the allowables as follows:

$$\frac{B_1 P_{\max} D_o}{2t} + \frac{B_2(M_A + M_B)}{Z} \leq kS_h \qquad \text{(B4.10)}$$

where M_B = resultant moment loading on cross section due to occasional loads, such as thrusts from relief and safety valves, loads from pressure and flow transients, and earthquake, if required. For earthquake, use only one-half the range. Effects of anchor displacement due to earthquake may be excluded if they are included under thermal expansion

$P_{\max}$ = peak pressure, psi

kS_h = $1.8S_h$ for service level B (upset condition) but not greater than $1.5S_y$; $2.25S_h$ for service level C (emergency condition) but not greater than $1.8S_y$; and $3.0S_h$ for service level D (faulted condition) but not greater than $2.0S_y$

S_h = material allowable stress at temperature consistent with loading under consideration, psi

S_y = material yield strength at temperature consistent with loading under consideration, psi

3. *Stresses due to thermal expansion*

a. Thermal expansion stress range must meet the allowable S_A, that is,

$$\frac{iM_C}{Z} \leq S_A \qquad \text{(B4.11)}$$

where S_A = allowable stress range for expansion stresses
= $f(1.25S_c + 0.25S_h)$, psi

f = stress range reduction factor, as in Table B4.2

M_c = range of resultant moment due to thermal expansion, in · lb; also include moment effects of anchor displacements due to earthquake if anchor displacement effects were omitted from occasional loadings

S_c = basic material allowable stress at minimum (cold) temperature, psi

TABLE B4.2 Stress-Range Reduction Factors

Number of equivalent full-temperature cycles N	f
7,000 and less	1.0
7,000 to 14,000	0.9
14,000 to 22,000	0.8
22,000 to 45,000	0.7
45,000 to 100,000	0.6
100,000 and over	0.5

Source: Table NC-3611.2(e)-1, Section III, Division 1, ASME B & PV Code, 1998. (*Courtesy of ASME.*)

S_h = basic material allowable stress at maximum (hot) temperature, psi
i = stress intensification factor

b. If Eq. (B4.11) is not met, the piping may be qualified by meeting the following equation:

$$\frac{PD_o}{4t} + \frac{0.75iM_A}{Z} + \frac{iM_C}{Z} \leq S_h + S_A \qquad \text{(B4.12)}$$

where $0.75i$ shall not be less than 1.0.

4. *Stresses due to nonrepeated anchor movement.* The effect of any single nonrepeated anchor movement (such as building settlement) must meet $3.0S_c$,

$$\frac{iM_D}{Z} \leq 3.0S_c \qquad \text{(B4.13)}$$

where M_D = resultant moment due to any single nonrepeated anchor movement (e.g., predicted building settlement), in · lb.

5. The *stress-intensification factor (SIF)* is defined as the ratio of the maximum stress intensity to the nominal stress, calculated by the ordinary formulas of mechanics. It is used as a safety factor to account for the effect of localized stresses on piping under a repetitive loading. In piping design, this factor is applied to welds, fittings, branch connections, and other piping components where stress concentrations and possible fatigue failure might occur. Usually, experimental methods are used to determine these factors.

It is recognized that some of the SIFs for the same components are different for different codes. In some cases, different editions of the same code provide different SIFs for a given component. The way that the SIFs are applied to moment loadings is also different for different codes. The B31.1 and ASME Section III codes require that the same SIF be applied to all the three-directional moments while the B31.3, B31.4, B31.5, and B31.8 codes require that different SIFs be applied to the in-plane and out-of-plane moments, with no SIF required for torsion (see Fig. B4.5*a* and figure note 10).

Therefore, the stress analyst has to ensure that the appropriate SIFs from the applicable code (i.e., committed code) are used. The formulas for SIFs in the ASME Section III code (1989 edition) are given in Fig. B4.5c for reference.

Recommended SIFs for some piping components which are not addressed in the code are listed below:

a. Weldolets or sockolets[4]

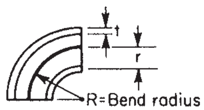
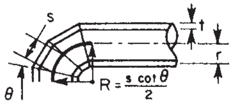
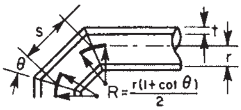
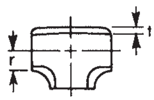
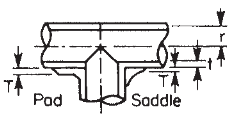
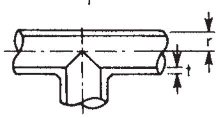
(1) If $r/R > 0.5$,

$$i = \frac{0.9}{h^{2/3}} \qquad \text{where} \qquad h = \frac{3.3t}{R} \qquad \text{(B4.14)}$$

where r = mean radius of branch pipe, in
t = wall thickness of run pipe, in
R = mean radius of run pipe, in

Item	Description	Flexibility factor n	Stress int. factor i [Notes (9), (10)]	
			In-plane i	Out-of-plane i_o
1	Welding elbow[1,2,3,5,8] or pipe bend	$\frac{1.65}{h}$	$\frac{0.9}{h^{2/3}}$	$\frac{0.75}{h^{2/3}}$
2	Closely spaced miter bend[1,2,3,8] $s < r(1 + \tan\theta)$	$\frac{1.52}{h^{5/6}}$	$\frac{0.9}{h^{2/3}}$	$\frac{0.75}{h^{2/3}}$
3	Widely spaced miter bend,[1,2,4,8] $s \geqq r(1 + \tan\theta)$	$\frac{1.52}{h^{5/6}}$	$\frac{0.9}{h^{2/3}}$	$\frac{0.75}{h^{2/3}}$
4	Welding tee[1,2] per ANSI B16.9	1	$\frac{3}{4}i_o + \frac{1}{4}$	$\frac{0.9}{h^{2/3}}$
5	Reinforced fabricated tee,[1,2,6] with pad or saddle	1	$\frac{3}{4}i_o + \frac{1}{4}$	$\frac{0.9}{h^{2/3}}$
6	Unreinforced fabricated tee[1,2,6]	1	$\frac{3}{4}i_o + \frac{1}{4}$	$\frac{0.9}{h^{2/3}}$
7	Butt-welded joint, reducer, or welding neck flange	1	1.0	
8	Double-welded slip-on flange	1	1.2	
9	Fillet-welded joint, or socket weld flange	1	1.3	
10	Lap-joint flange (with ANSI B16.9 lap-joint stub)	1	1.6	
11	Threaded pipe joint, or threaded flange	1	2.3	
12	Corrugated straight pipe, or corrugated or creased bend[7]	5	2.5	

FIGURE B4.5*a* Flexibility factor n and stress intensification factors i_i and i_o per ASME B31.3, B31.4, B31.5, and B31.8 codes.

Flexibility characteristic h	Sketch
$\frac{tR}{r^2}$	
$\frac{\cot\theta}{2}\frac{ts}{r^2}$	
$\frac{1+\cot\theta}{2}\frac{t}{r}$	
$4.4\frac{t}{r}$	
$\frac{(t+\frac{1}{2}T)^{5/2}}{t^{3/2}r}$	
$\frac{t}{r}$	

Meaning of symbols:

r = mean radius of matching pipe
t = (for elbows, bends, and miter bends) nominal wall thickness of elbow, bend, or miter bend; see Note 5
t = (for tees and nozzles), the nominal wall thickness of the matching pipe
R = bend radius of pipe bend or elbow
θ = one-half angle between adjacent miter axes
s = miter spacing at center line
T = pad or saddle thickness, in.

NOTES:

1. The flexibility factors n and the stress-intensification factors i in the table apply to bending and in no case shall be taken less than unity. Factors for torsion equal unity. Both factors apply over the effective arc length (shown by heavy center lines in the sketches) for curved and miter elbows and to the intersection point for tees.
2. Those flexibility and stress intensification factors which are proportional to a power of the characteristic h may be read from the graph of Fig. B4-5*b*.
3. Where flanges are attached at one or both ends, the values of n and i in the table shall be corrected by the multiplicative factors C_f given by the formulas:

$$C_f = h^{1/6} \text{ (one end flanged)}$$
$$C_f = h^{1/3} \text{ (both ends flanged)}$$

4. Also includes single-miter joint.
5. Cast butt-welding elbows may have considerably heavier walls than that of the pipe with which they are used. Large errors may be introduced unless the effect of such greater thickness is considered.
6. $h = 4.05t/r$ for $T > 1\frac{1}{2}t$
7. Factors shown apply to bending; flexibility factor for torsion = 0.9.
8. In large-diameter thin-wall elbows and bends internal pressure can significantly decrease both flexibility and stress intensification factors. To correct values obtained from above tabulation so as to account for internal pressure, divide the flexibility factor by the quantity

$$[1 + 6(P/E)(r/t)^{7/3}(R/r)^{1/3}]$$

and divide the stress intensification factor by the quantity

$$[1 + 3.25(P/E)(r/t)^{5/2}(R/r)^{2/3}]$$

where P = internal gage pressure and E = Young's modulus.

9. For items 2 and 3, $i_o = 0.9/h^{2/3}$ per ASME B31.3 and B31.8 codes.
10. For ASME B31.3 and B31.8 piping, a single intensification factor equal to $0.9/h^{2/3}$ may be used for both i_i and i_o if desired.

FIGURE B4.5*a* (*Continued*) Flexibility factor n and stress intensification factors i_i and i_o per ASME B31.3, B31.4, B31.5, and B31.8 codes.

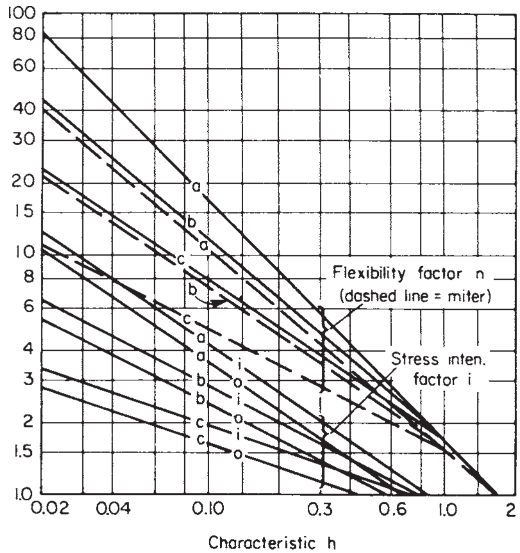


FIGURE B4.5*b* Flexibility and stress intensification factors for those cases where result is a power of the characteristic *h*, defined in Fig. B4.5*a*. Key: *a* = no flanges; *b* = one flange; *c* = two flanges; *i* = in plane; *o* = out of plane.

(2) If $r/R \leq 0.5$,

$$i = 1.5\left(\frac{R}{T}\right)^{2/3}\left(\frac{r}{R}\right)^{1/2}\frac{t}{T}\frac{r}{r_p} \qquad \text{(B4.15)}$$

or

$$i = \frac{0.9}{(3.3T/R)^{2/3}} \qquad \text{(B4.16)}$$

whichever is less and

where R = run pipe mean radius, in
T = run pipe wall thickness, in
r = branch pipe mean radius, in
t = branch pipe wall thickness, in
r_p = outer radius of weldolet, in

SIF values for typical weldolet branch connections with $r/R \leq 0.5$ are tabulated in Tables B4.3*a* to B4.3*l*.

b. *Half-Couplings (Welding Boss)*. For half-couplings with $r/R \leq 0.5$, use the above branch connection Eq. (B4.15) or the unreinforced fabricated tee equation, whichever is less. For half-coupling with $r/R > 0.5$, use the unreinforced fabricated tee formula. Tables B4.4*a* to B4.4*f* give SIFs for commonly used half-coupling

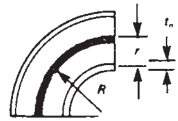
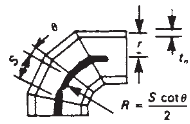
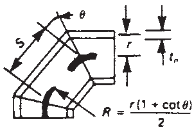
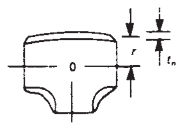
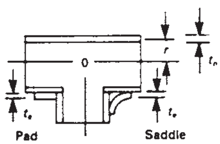
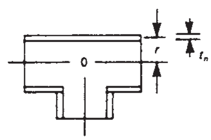
Description	Flexibility Characteristic h	Flexibility Factor k	Stress Intensification Factor i	Sketch
Welding elbow or pipe bend [Notes (1), (2), (3)]	$\frac{t_n R}{r^2}$	$\frac{1.65}{h}$	$\frac{0.9}{h^{2/3}}$	
Closely spaced miter bend [Notes (1), (2), (3)] $s < r(1 + \tan\theta)$	$\frac{s t_n \cot\theta}{2r^2}$	$\frac{1.52}{h^{5/6}}$	$\frac{0.9}{h^{2/3}}$	
Widely spaced miter bend [Notes (1), (2), (4)] $s \geq r(1 + \tan\theta)$	$\frac{t_n(1 + \cot\theta)}{2r}$	$\frac{1.52}{h^{5/6}}$	$\frac{0.9}{h^{2/3}}$	
Welding tee per ANSI B16.9 [Notes (1), (2)]	$\frac{4.4\ t_n}{r}$	1	$\frac{0.9}{h^{2/3}}$	
Reinforced fabricated tee [Notes (1), (5), (10)]	$\frac{\left(t_n + \frac{t_e}{2}\right)^{5/2}}{r(t_n)^{3/2}}$	1	$\frac{0.9}{h^{2/3}}$	
Unreinforced fabricated tee [Notes (1), (10)]	$\frac{t_n}{r}$	1	$\frac{0.9}{h^{2/3}}$	

FIGURE B4.5*c* Flexibility and stress intensification factors ($D_o/t_n \leq 100$). (*Figure NC-3673.2(b)-1, Section III, Division 1, ASME B & PV Code, 1998. Courtesy of ASME.*)

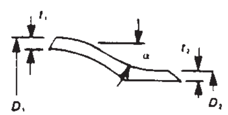
Description	Flexibility Factor k	Stress Intensification Factor i	Sketch
Branch connection [Note (6)]	1	For checking branch end $Z = \pi (r'_m)^2 T'_b$ $i = 1.5 \left(\frac{R_m}{T_r}\right)^{2/3} \left(\frac{r'_m}{R_m}\right)^{1/2} \left(\frac{T'_b}{T_r}\right) \left(\frac{r'_m}{r_p}\right)$ For checking run ends $Z = \pi (R_m)^2 T_r$ $i = 0.4 \left(\frac{R_m}{T_r}\right)^{2/3} \left(\frac{r'_m}{R_m}\right)$ but not less than 1.5	Fig. NC-3673.2(b)-2
Girth butt weld [Note (1)] $t_n \geq 0.237$ in.	1	1.0	
Girth butt weld [Note (1)] $t_n < 0.237$ in.	1	1.9 max. or $0.9(1 + 3\delta/t_n)$ but not less than 1.0	
Circumferential fillet welded or socket welded joints [Note (11)]	1	$2.1/(C_x/t_n)$ but not less than 1.3	Fig. NC-4427-1 sketches (c-1), (c-2), and (c-3)
Brazed joint	1	2.1	Fig. NC-4511-1
30 deg. tapered transition (ANSI B16.25) [Note (1)]	1	1.9 max. or $1.3 + 0.0036 \frac{D_o}{t_n} + 3.6 \frac{\delta}{t_n}$	
Concentric and eccentric reducers [Note (7)] (ANSI B16.9)	1	2.0 max. or $0.5 + 0.01\alpha \left(\frac{D_2}{t_2}\right)^{1/2}$	
Threaded pipe joint or threaded flange	1	2.3	
Corrugated straight pipe or corrugated or creased bend [Note (8)]	5	2.5	

FIGURE B4.5*c* (*Continued*) Flexibility and stress intensification factors ($D_o/t_n \leq 100$). (*Figure NC-3673.2(b)-1, Section III, Division 1, ASME B & PV Code, 1998. Courtesy of ASME.*)

configurations. If the half-coupling rating is not known, assume a Class 3000 half-coupling, since this will give the more conservative value.

c. *Sweepolets*. For branch:

$$i = 0.45 \left(\frac{R}{T}\right)^{2/3} \left(\frac{r}{R}\right)^{1/2} \frac{t}{T} F_1 F_s \tag{B4.17}$$

(1) The following nomenclature applies:
r = mean radius of pipe, in. (matching pipe for tees and elbows)
t_n = nominal wall thickness of pipe in. [matching pipe for tees and elbows, see Note (9)]
R = bend radius of elbow or pipe bend, in.
θ = one-half angle between adjacent miter axes, deg.
s = miter spacing at center line, in.
t_e = reinforced thickness, in.
δ = average permissible mismatch at girth butt welds as shown in Fig. NC-4233-1. A value of δ less than $\frac{1}{32}$ in. may be used provided the smaller mismatch is specified for fabrication. For "flush" welds, as defined in Fig. NB-3683.1(c)-1, δ may be taken as zero, i = 1.0, and flush welds need not be ground.
D_o = outside diameter, in.

(2) The flexibility factors k and stress intensification factors i apply to bending in any plane for fittings and shall in no case be taken less than unity. Both factors apply over the effective arc length (shown by heavy center lines in the sketches) for curved and miter elbows, and to the intersection point for tees. The values of k and i can be read directly by entering with the characteristic h computed from the equations given.

(3) Where flanges are attached to one or both ends, the values of k and i shall be corrected by the factor c given below.
(a) One end flanged, $c = h^{1/6}$
(b) Both ends flanged, $c = h^{1/3}$

(4) Also includes single miter joints.

(5) When $t_e > 1.5t_n$, $h = 4.05t_n/r$

(6) The equation applies only if the following conditions are met.
(a) The reinforcement area requirements of NC-3643 are met.
(b) The axis of the branch pipe is normal to the surface of run pipe wall.
(c) For branch connections in a pipe, the arc distance measured between the centers of adjacent branches along the surface of the run pipe is not less than three times the sum of their inside radii in the longitudinal direction or not less than two times the sum of their inside radii along the circumference of the run pipe.
(d) The inside corner radius r_1 [Fig. NC-3673.2(b)-2] for nominal branch pipe size greater than 4 in. shall be between 10% and 50% T_r. The radius r_1 is not required for nominal branch pipe size smaller than 4 in.
(e) The outer radius r_2 is not less than the larger of $T_b/2$, $(T_{b'} + Y)/2$ [Fig. NC-3673-2(b)-2 sketch (c)] or $T_r/2$.
(f) The outer radius r_3 is not less than the larger of
(1) $0.002\theta\ d_o$
(2) 2 $(\sin \theta)^3$ times the offset for the configurations shown in Fig. NC-3673.2(b)-2 sketches (a) and (b).
(g) $R_m/T_r \leq 50$ and $r'_m/R_m \leq 0.5$.
(h) The outer radius r_2 is not required provided an additional multiplier of 2.0 is included in the equations for branch end and run end stress intensification factors. In this case, the calculated value of i for the branch or run shall not be less than 2.1.

(7) The equation applies only if the following conditions are met.
(a) Cone angle α does not exceed 60 deg.
(b) The larger of D_1/t_1 and D_2/t_2 does not exceed 100.
(c) The wall thickness is not less than t_1 throughout the body of the reducer, except in and immediately adjacent to the cylindrical portion on the small end, where the thickness shall not be less than t_2.
(d) For eccentric reducers, α is the maximum cone angle.

(8) Factors shown apply to bending; flexibility factor for torsion equals 0.9.

(9) The designer is cautioned that cast butt welding elbows may have considerably heavier walls than that of the pipe with which they are used. Large errors may be introduced unless the effect of these greater thicknesses is considered.

(10) The stress intensification factor i shall in no case be taken as less than 2.1.

(11) In Fig. NC-4427-1(c-1) and (c-2), C_x shall be taken as X_{min} and $C_x \geq 1.25\ t_n$. In Fig. NC-4427-1 (c-3), $C_x \geq 0.75\ t_n$. For unequal leg lengths use the smaller leg length for C_x.

FIGURE B4.5*c* (*Continued*) Flexibility and stress intensification factors ($D_o/t_n \leq 100$). (*Figure NC-3673.2(b)-1, Section III, Division 1, ASME B & PV Code, 1998. Courtesy of ASME.*)

For run:

$$\text{For } \frac{r}{R} > 0.5 \qquad i = 0.40\left(\frac{R}{T}\right)^{2/3}(F_2)(F_s) \tag{B4.18}$$

$$\text{For } \frac{r}{R} \leq 0.5 \qquad i = 0.8\left(\frac{R}{T}\right)^{2/3}\left(\frac{r}{R}\right)F_s \quad \text{but not less than } 1.5 \tag{B4.19}$$

where F_1 = F_2 = 1.0 for flush or dressed insert welds
= 1.6 for as-welded insert welds
F_2 = $(0.5 + r/R)$, but not less than 1.0 for as-welded insert welds
F_s = $1 + 0.05(r - 3)$, but not less than 1.0
R = mean radius of run pipe, in

TABLE B4.3*a* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 1½ or DN 40)

Branch pipe			Run pipe, NPS 1½ (DN 40)			
			Run pipe schedule			
			10*s*	40	80	160
			Run pipe wall thickness			
NPS	Sch.	Thickness	0.109"	0.145"	0.200"	0.281"
½ (DN 15)	10*s*	0.083"	1.665	1.031	1.000	—
	40	0.109"	2.074	1.285	1.000	1.000
	80	0.147"	2.582	1.599	1.000	1.000
	160	0.188"	—	—	1.086	2.000

1 in = 25.4 mm

r = mean radius of branch pipe, in
T = nominal wall thickness of run pipe, in
t = nominal wall thickness of branch pipe, in

If a more detailed analysis is desirable, see Ref. 5 for the equations to be used for moment separation.

d. Lateral branch connections (45°).[6,7] For $r_b/r > 0.5$,

$$i = \frac{0.9}{h^{2/3}} \qquad \text{where} \qquad h = \frac{1.97t}{r} \tag{B4.20}$$

TABLE B4.3*b* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 2 or DN 50)

Branch pipe			Run pipe, NPS 2 (DN 50)			
			Run pipe schedule			
			10*s*	40	80	160
			Run pipe wall thickness			
NPS	Sch.	Thickness	0.109"	0.154"	0.218"	0.344"
½ (DN 15)	10*s*	0.083"	1.731			
	40	0.109"	2.157	1.209	1.000	
	80	0.147"			1.000	1.000
	160	0.188"			1.000	1.000
¾ (DN 20)	10*s*	0.083"	1.964			
	40	0.113"	2.550	1.429	1.000	
	80	0.154"			1.015	1.000
	160	0.219"			1.286	1.000

1 in = 25.4 mm

TABLE B4.3c SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 2½ (DN 65) and NPS 3 (DN 80)

Branch pipe			Run pipe, NPS 2½ (DN 65)				Run pipe, NPS 3 (DN 80)			
			Run pipe schedule				Run pipe schedule			
			10s	40	80	160	10s	40	80	160
			Run pipe wall thickness				Run pipe wall thickness			
NPS	Sch.	Thickness	0.120"	0.203"	0.276"	0.375"	0.120"	0.216"	0.300"	0.438"
½ (DN 15)	10s	0.083"	1.524	1.000			1.576	1.000		
	40	0.109"	1.899	1.000	1.000		1.965	1.000	1.000	
	80	0.147"		1.000	1.000	1.000		1.000	1.000	1.000
	160	0.188"			1.000	1.000			1.000	1.000
¾ (DN 20)	10s	0.083"	1.728	1.000			1.788	1.000		
	40	0.113"	2.245	1.000	1.000		2.322	1.000	1.000	
	80	0.154"		1.185	1.000	1.000		1.106	1.000	1.000
	160	0.219"			1.000	1.000			1.000	1.000
1 (DN 25)	10s	0.109"	2.603	1.078			2.694	1.006		
	40	0.133"	3.082	1.277	1.000		3.189	1.192	1.000	
	80	0.179"		1.619	1.000	1.000		1.511	1.000	1.000
	160	0.250"			1.225	1.000			1.103	1.000

1 in = 25.4 mm

TABLE B4.3*d* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 4 (DN 100) and 6 or (DN 150)

Branch pipe			Run pipe, NPS 4 (DN 100)					Run pipe, NPS 6 (DN 150)				
			Run pipe schedule					Run pipe schedule				
			10*s*	40	80	120	160	10*s*	40	80	120	160
			Run pipe wall thickness					Run pipe wall thickness				
NPS (DN)	Sch.	Thickness	0.120"	0.237"	0.337"	0.438"	0.531"	0.134"	0.280"	0.432"	0.562"	0.719"
½ (15)	10*s*	0.083"	1.646	1.000				1.462	1.000			
	40	0.109"	2.051	1.000	1.000			1.822	1.000	1.000		
	80	0.147"		1.000	1.000	1.000			1.000	1.000	1.000	
	160	0.188"			1.000	1.000	1.000			1.000	1.000	1.000
¾ (20)	10*s*	0.083"	1.867	1.000				1.659	1.000			
	40	0.113"	2.425	1.000	1.000			2.154	1.000	1.000		
	80	0.154"		1.109	1.000	1.000			1.000	1.000	1.000	
	160	0.219"			1.000	1.000	1.000			1.000	1.000	1.000
1 (25)	10*s*	0.109"	2.813	1.000				2.499	1.000			
	40	0.133"	3.330	1.066	1.000			2.958	1.000	1.000		
	80	0.179"		1.352	1.000	1.000			1.094	1.000	1.000	
	160	0.250"			1.009	1.000	1.000			1.000	1.000	1.000
1½ (40)	10*s*	0.109"	3.762	1.205				3.342	1.000			
	40	0.145"	4.855	1.555	1.000			4.313	1.258	1.000		
	80	0.200"		2.044	1.132	1.000			1.654	1.000	1.000	
	160	0.281"			1.548	1.000	1.000			1.092	1.000	1.000
2 (50)	10*s*	0.109"						3.907	1.140			
	40	0.154"						5.357	1.563	1.000		
	80	0.218"							2.117	1.023	1.000	
	160	0.344"								1.620	1.041	1.000
2½ (65)	10*s*	0.120"						4.968	1.449			
	40	0.203"						8.027	2.342	1.132		
	80	0.276"							3.054	1.477	1.000	
	160	0.375"								2.017	1.296	1.000

1 in = 25.4 mm

TABLE B4.3*e* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 8 or DN 200)

Branch pipe			Run pipe, NPS 8 (DN 200)				
			Run pipe schedule				
			10s	40	80	120	160
			Run pipe wall thickness				
NPS (DN)	Sch.	Thickness	0.148"	0.322"	0.500"	0.719"	0.906"
½ (15)	10s	0.083"	1.295	1.000			
	40	0.109"	1.614	1.000	1.000		
	80	0.147"		1.000	1.000		
	160	0.188"			1.000	1.000	1.000
¾ (20)	10s	0.083"	1.470	1.000			
	40	0.113"	1.909	1.000	1.000		
	80	0.154"		1.000	1.000		
	160	0.219"			1.000	1.000	1.000
1 (25)	10s	0.109"	2.214	1.000			
	40	0.133"	2.621	1.000	1.000		
	80	0.179"		1.000	1.000		
	160	0.250"			1.000	1.000	1.000
1½ (40)	10s	0.109"	2.961	1.000			
	40	0.145"	3.821	1.042	1.000		
	80	0.200"		1.371	1.000		
	160	0.281"			1.000	1.000	1.000
2 (50)	10s	0.109"	3.461	1.000			
	40	0.154"	4.746	1.295	1.000		
	80	0.218"		1.754	1.000		
	160	0.344"			1.329	1.000	1.000
2½ (65)	10s	0.120"	4.401	1.201			
	40	0.203"	7.112	1.940	1.000		
	80	0.276"		2.530	1.211		
	160	0.375"			1.654	1.000	1.000
3 (80)	10s	0.120"	5.049	1.377			
	40	0.216"	8.704	2.374	1.136		
	80	0.300"		3.172	1.518		
	160				2.102	1.142	1.000

1 in = 25.4 mm

$$\text{For } \frac{r_b}{r} \leq 0.5 \quad i = 0.537\left(\frac{r}{t}\right)^{2/3}\left(\frac{r_b}{r}\right)^{1/2} \tag{B4.21}$$

where t = run pipe wall thickness, in
r = run pipe mean radius, in
r_b = branch pipe mean radius, in

These equations are for integrally reinforced branch connections such as latrolets. By the analogy used in Fig. NC-3673.2(b)-1 in Section III of the ASME Code, the SIF for unreinforced 45° branch connections (stub-ins) can be obtained by multiplying the factors obtained above by $(4.4)^{2/3} = 2.685$.

TABLE B4.3f SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 10 (DN 250))

Branch pipe			Run pipe, NPS 10 (DN 250)				
			Run pipe schedule				
			10*s*	40	80	120	160
			Run pipe wall thickness				
NPS (DN)	Sch.	Thickness	0.165"	0.365"	0.594"	0.844"	1.125"
¾ (20)	10*s*	0.083"	1.272	1.000			
	40	0.113"	1.652	1.000	1.000		
	80	0.154"		1.000	1.000		
	160	0.219"			1.000	1.000	1.000
1 (25)	10*s*	0.109"	1.916	1.000			
	40	0.133"	2.269	1.000	1.000		
	80	0.179"		1.000	1.000		
	160	0.250"			1.000	1.000	1.000
1½ (40)	10*s*	0.109"	2.563	1.000			
	40	0.145"	3.308	1.000	1.000		
	80	0.200"		1.154	1.000		
	160	0.281"			1.000	1.000	1.000
2 (50)	10*s*	0.109"	2.997	1.000			
	40	0.154"	4.108	1.090	1.000		
	80	0.218"		1.477	1.000		
	160	0.344"			1.035	1.000	1.000
2½ (65)	10*s*	0.120"	3.810	1.011			
	40	0.203"	6.157	1.634	1.000		
	80	0.276"		2.131	1.000		
	160	0.375"			1.288	1.000	1.000
3 (80)	10*s*	0.120"	4.371	1.160			
	40	0.216"	7.535	2.000	1.000		
	80	0.300"		2.672	1.182		
	160	0.438"			1.637	1.000	1.000
4 (100)	10*s*	0.120"	5.172	1.373			
	40	0.237"	9.807	2.603	1.152		
	80	0.337"		3.572	1.581		
	120	0.438"			1.980	1.098	1.000
	160	0.531"			2.318	1.286	1.000

1 in = 25.4 mm

e. Pipet

$$i = \frac{0.9}{h^{2/3}} \tag{B4.22}$$

For butt-weld pipet $h = \dfrac{3.3t}{r}$

For socket-weld pipet $h = \dfrac{5.9t}{r}$

For threaded pipet $h = \dfrac{4.19t}{r}$

TABLE B4.3*g* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 12 or DN 300)

Branch pipe			Run pipe, NPS 12 (DN 300)					
			Run pipe schedule					
			10*s*	40*s*	40	80	120	160
			Run pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.180"	0.375"	0.406"	0.688"	1.000"	1.312"
¾ (20)	10*s*	0.083"	1.132	1.000	1.000			
	40	0.113"	1.471	1.000	1.000			
	80	0.154"		1.000	1.000	1.000		
	160	0.219"				1.000	1.000	1.000
1 (25)	10*s*	0.109"	1.706	1.000	1.000			
	40	0.133"	2.020	1.000	1.000			
	80	0.179"		1.000	1.000	1.000	1.000	1.000
	160	0.250"				1.000	1.000	1.000
1½ (40)	10*s*	0.109"	2.282	1.000	1.000			
	40	0.145"	2.944	1.000	1.000			
	80	0.200"		1.136	1.000	1.000	1.000	1.000
	160	0.281"				1.000	1.000	1.000
2 (50)	10*s*	0.109"	2.667		1.000			
	40	0.154"	3.657	1.073	1.000			
	80	0.218"		1.454	1.273	1.000	1.000	1.000
	160	0.344"				1.000	1.000	1.000
2½ (65)	10*s*	0.120"	3.392		1.000			
	40	0.203"	5.480	1.609	1.409			
	80	0.276"		2.098	1.837	1.000	1.000	1.000
	160	0.375"				1.038	1.000	1.000
3 (80)	10*s*	0.120"	3.890		1.000			
	40	0.216"	6.707	1.969	1.724			
	80	0.300"		2.630	2.303	1.000	1.000	1.000
	160	0.438"				1.319	1.000	1.000
4 (100)	10*s*	0.120"	4.603		1.183			
	40	0.237"	8.730	2.562	2.244			
	80	0.337"		3.516	3.079	1.273	1.000	1.000
	120	0.438"				1.595	1.000	1.000
	160	0.531"				1.868	1.000	1.000

1 in = 25.4 mm

f. Branchlet

$$i = \frac{0.9}{h^{2/3}} \quad \text{where} \quad h = \frac{3.8t}{r} \tag{B4.23}$$

g. Reducing elbow

$$i = \frac{0.9}{(TB/R^2)^{2/3}} \quad \text{or} \quad 2.0\text{—whichever is higher} \tag{B4.24}$$

TABLE B4.3*h* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 14 or DN 350)

Branch pipe			Run pipe, NPS 14 (DN 350)					
			Run pipe schedule					
			10	30	40	80	120	160
			Run pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.250"	0.375"	0.438"	0.750"	1.094"	1.406"
1	10*s*	0.109"	1.002	1.000				
(25)	40	0.133"	1.186	1.000	1.000			
	80	0.179"		1.000	1.000	1.000		
	160	0.250"				1.000	1.000	1.000
1½	10*s*	0.109"	1.340	1.000				
(40)	40	0.145"	1.729	1.000	1.000			
	80	0.200"		1.155	1.000	1.000		
	160	0.281"				1.000	1.000	1.000
2	10*s*	0.109"	1.566	1.000				
(50)	40	0.154"	2.147	1.091	1.000			
	80	0.218"		1.478	1.140	1.000		
	160	0.344"				1.000	1.000	1.000
2½	10*s*	0.120"	1.991	1.012				
(65)	40	0.203"	3.218	1.635	1.261			
	80	0.276"		2.132	1.644	1.000		
	160	0.375"				1.000	1.000	1.000
3	10*s*	0.120"	2.284	1.160				
(80)	40	0.216"	3.938	2.000	1.543			
	80	0.300"		2.673	2.062	1.000		
	160	0.438"				1.160	1.000	1.000
4	10*s*	0.120"	2.703	1.373				
(100)	40	0.237"	5.126	2.604	2.009			
	80	0.337"		3.573	2.756	1.120		
	120	0.438"				1.403	1.000	1.000
	160	0.531"				1.643	1.000	1.000
6	10*s*	0.134"	3.681	1.870				
(150)	40	0.280"	7.434	3.776	2.913			
	80	0.432"		5.618	4.334	1.761		
	120	0.562"				2.268	1.203	1.000
	160	0.719"				2.789	1.480	1.000

1 in=25.4 mm

where T = wall thickness of large end, in
B = actual bend radius, in
R = mean radius of large end, in

ASME B31.1 Power Piping Code.[8] This code concerns nonnuclear piping such as that found in the turbine building of a nuclear plant or in a fossil-fuel power plant. Piping services include steam, water, oil, gas, and air. Design requirements of this

TABLE B4.3*i* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 16 or DN 400)

Branch pipe			Run pipe, NPS 16 (DN 400)					
			Run pipe schedule					
			10	30	40	80	120	160
			Run pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.250"	0.375"	0.500"	0.844"	1.219"	1.594"
1	10*s*	0.109"	1.025	1.000				
(25)	40	0.133"	1.213	1.000	1.000			
	80	0.179"		1.000	1.000			
	160	0.250"				1.000	1.000	1.000
1½	10*s*	0.109"	1.370	1.000				
(40)	40	0.145"	1.768	1.000	1.000			
	80	0.200"		1.181	1.000	1.000		
	160	0.281"				1.000	1.000	1.000
2	10*s*	0.109"	1.602	1.000				
(50)	40	0.154"	2.196	1.116	1.000			
	80	0.218"		1.512	1.000	1.000		
	160	0.344"				1.000	1.000	1.000
2½	10*s*	0.120"	2.037	1.035				
(65)	40	0.203"	3.291	1.672	1.034			
	80	0.276"		2.181	1.349	1.000		
	160	0.375"				1.000	1.000	1.000
3	10*s*	0.120"	2.337	1.187				
(80)	40	0.216"	4.028	2.047	1.265			
	80	0.300"		2.734	1.691	1.000		
	160	0.438"				1.000	1.000	1.000
4	10*s*	0.120"	2.765	1.405				
(100)	40	0.237"	5.243	2.664	1.647			
	80	0.337"		3.656	2.260	1.000		
	120	0.438"				1.179	1.000	1.000
	160	0.531"				1.380	1.000	1.000
6	10*s*	0.134"	3.766	1.913				
(150)	40	0.280"	7.605	3.864	2.389			
	80	0.432"		5.748	3.554	1.480		
	120	0.562"				1.905	1.028	1.000
	160	0.719"				2.343	1.264	1.000

1 in = 25.4 mm

code cover those for pipe, flanges, bolting, gaskets, valves, relief devices, fittings, and the pressure-containing portions of other piping components. It also includes hangers and supports and other equipment items necessary to prevent overstressing the pressure-containing components.

The loadings required to be considered are pressure; weight (live, dead, and under test loads); impact (e.g., water hammer); wind; earthquake (where applicable); vibration; and those loadings resulting from thermal expansion and contraction.

TABLE B4.3*j* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 18 or DN 450)

Branch pipe			Run pipe, NPS 18 (DN 450)					
			Run pipe schedule					
			10	STD	XS	40	80	120
			Run pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.250"	0.375"	0.500"	0.562"	1.938"	1.375"
1½	10*s*	0.109"	1.398	1.000				
(65)	40	0.145"	1.804	1.000	1.000			
	80	0.200"		1.205	1.000	1.000	1.000	1.000
	160	0.281"			1.018	1.000	1.000	1.000
2	10*s*	0.109"	1.634	1.000				
(50)	40	0.154"	2.241	1.139	1.000			
	80	0.218"		1.543	1.000	1.000	1.000	1.000
	160	0.344"			1.510	1.242	1.000	1.000
2½	10*s*	0.120"	2.078	1.056				
(65)	40	0.203"	3.358	1.706	1.055			
	80	0.276"		2.225	1.376	1.132	1.000	1.000
	160	0.375"			1.880	1.546	1.000	1.000
3	10*s*	0.120"	2.384	1.211				
(80)	40	0.216"	4.109	2.088	1.291			
	80	0.300"		2.790	1.725	1.419	1.000	1.000
	160	0.438"			2.388	1.965	1.000	1.000
4	10*s*	0.120"	2.821	1.433				
(100)	40	0.237"	5.349	2.718	1.681			
	80	0.337"		3.730	2.306	1.897	1.000	1.000
	120	0.438"			2.889	2.376	1.008	1.000
	160	0.531"			3.383	2.782	1.181	1.000
6	10*s*	0.134"	3.841	1.952				
(150)	40	0.280"	7.758	3.942	2.438			
	80	0.432"		5.865	3.627	2.983	1.266	1.000
	120	0.562"			4.669	3.840	1.629	1.000
	160	0.719"			5.743	4.723	2.004	1.055
8	10*s*	0.148"	4.887	2.483				
(200)	40	0.322"	8.106	5.237	3.239			
	80	0.500"			4.868	4.004	1.699	1.000
	120	0.719"			6.907	5.681	2.410	1.269
	160	0.906"			8.396	6.906	2.930	1.542

1 in = 25.4 mm

The design equations and stress limits are as follows (terms are the same as those for Class 2 and 3 piping except for those defined below):

1. *Stress due to sustained loads.* The effects of pressure, weight, and other sustained mechanical loads must meet the requirements of Eq. (B4.25):

$$S_L = \frac{PD_o}{4t} + \frac{0.75iM_A}{Z} \leq 1.0S_h \qquad \text{(B4.25)}$$

TABLE B4.3*k* SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 20 or DN 500)

Branch pipe			Run pipe, NPS 20 (DN 500)					
			Run pipe schedule					
			10*s*	20	XS	40	80	120
			Run pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.218"	0.375"	0.500"	0.594"	1.031"	1.500"
1½	10*s*	0.109"	1.789	1.000				
(40)	40	0.145"	2.308	1.000	1.000			
	80	0.200"		1.227	1.000	1.000	1.000	1.000
	160	0.281"		1.675	1.036	1.000	1.000	1.000
2	10*s*	0.109"	2.091	1.000				
(50)	40	0.154"	2.867	1.159	1.000	1.000		
	80	0.218"		1.570	1.000	1.000	1.000	1.000
	160	0.344"			1.538	1.156	1.000	1.000
2½	10*s*	0.120"	2.659	1.075				
(65)	40	0.203"	4.296	1.737	1.074			
	80	0.276"		2.266	1.401	1.054	1.000	1.000
	160	0.375"			1.914	1.439	1.000	1.000
3	10*s*	0.120"	3.050	1.233				
(80)	40	0.216"	5.257	2.126	1.315			
	80	0.300"		2.840	1.756	1.321	1.000	1.000
	160	0.438"			2.432	1.829	1.000	1.000
4	10*s*	0.120"	3.608	1.459				
(100)	40	0.237"	6.843	2.767	1.711			
	80	0.337"		3.797	2.348	1.766	1.000	1.000
	120	0.438"			2.942	2.212	1.000	1.000
	160	0.531"			3.445	2.590	1.026	1.000
6	10*s*	0.134"	4.915	1.987				
(150)	40	0.280"	9.925	4.013	2.482			
	80	0.432"		5.971	3.693	2.777	1.100	1.000
	120	0.562"			4.754	3.575	1.417	1.000
	160	0.719"				4.397	1.742	1.000
8	10*s*	0.148"	6.252	2.528				
(200)	40	0.322"		5.332	3.297			
	80	0.500"			4.957	3.727	1.477	1.000
	120	0.719"					2.095	1.117
	160	0.906"					2.547	1.358

1 in = 25.4 mm

where S_L = sum of longitudinal stresses due to pressure, weight, and other sustained loads, psi.

2. *Stress due to occasional loads.* The effects of pressure, weight, other sustained loads, and occasional loads including earthquake must meet the requirements of Eq. (B4.26):

$$\frac{PD_o}{4t} + \frac{0.75i(M_A + M_B)}{Z} \leq kS_h \tag{B4.26}$$

TABLE B4.3 / SIFs for Typical Weldolet Branch Connections (Run Pipe Size: NPS 24 or DN 600)

Branch pipe			Run pipe, NPS 24 (DN 600)					
			Run pipe schedule					
			10*s*	20	XS	40	80	120
			Run pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.218"	0.375"	0.500"	0.594"	1.031"	1.500"
2	10*s*	0.109"	1.716	1.000				
(50)	40	0.154"	2.352	1.196	1.000			
	80	0.218"		1.620	1.002	1.000	1.000	1.000
	160	0.344"			1.586	1.000	1.000	1.000
2½	10*s*	0.120"	2.181	1.109				
(65)	40	0.203"	3.525	1.792	1.108			
	80	0.276"		2.337	1.445	1.000	1.000	1.000
	160	0.375"			1.974	1.161	1.000	1.000
3	10*s*	0.120"	2.502	1.272				
(80)	40	0.216"	4.314	2.193	1.356			
	80	0.300"		2.929	1.812	1.066	1.000	1.000
	160	0.438"			2.509	1.475	1.000	1.000
4	10*s*	0.120"	2.961	1.505				
(100)	40	0.237"	5.615	2.854	1.765			
	80	0.337"		3.916	2.423	1.425	1.000	1.000
	120	0.438"			3.035	1.785	1.006	1.000
	160	0.531"			3.553	2.090	1.178	1.000
6	10*s*	0.134"	4.033	2.050				
(150)	40	0.280"	8.143	4.139	2.560			
	80	0.432"		6.158	3.809	2.240	1.262	1.000
	120	0.562"			4.904	2.884	1.625	1.106
	160	0.719"				3.547	1.999	1.361
8	10*s*	0.148"	5.130	2.608				
(200)	40	0.322"		5.499	3.402			
	80	0.500"			5.113	3.007	1.695	1.153
	120	0.719"				4.266	2.404	1.636
	160	0.906"				5.186	2.923	1.989
10	10*s*	0.165"	7.197	3.659				
(250)	40	0.365"		7.865	4.865			
	80	0.594"			7.645	4.496	2.534	1.724
	120	0.844"				6.375	3.593	2.445
	160	1.125"				8.148	4.591	3.125

1 in = 25.4 mm

where k = 1.15 for occasional loads acting less than 10 percent of operating period;
1.2 for occasional loads acting less than 1 percent of operating period

3. *Thermal expansion stress range*. The effects of thermal expansion must meet the requirements of Eq. (B4.27):

$$\frac{iM_C}{Z} \leq S_A + f(S_h - S_L) \tag{B4.27}$$

TABLE B4.4*a* SIFs for Class 3000 Half-Couplings (Branch Pipe Schedule 40) (Run Pipe Size: NPS 1½ to 8 (DN 40 to 200))

Run pipe			Sch. 40 branch pipe size and thickness					
			Branch pipe size					
			½ (15)	¾ (20)	1 (25)	1¼ (32)	1½ (40)	2 (50)
			Branch pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.109"	0.113"	0.133"	0.140"	0.145"	0.154"
1½ (40)	10*s*	0.109"	2.282	3.665	3.665	3.665	3.665	3.665
	40	0.145"	1.413	2.989	2.989	2.989	2.989	2.989
	80	0.200"	1.000	2.362	2.362	2.362	2.362	2.362
	160	0.281"	1.000	1.822	1.822	1.822	1.822	1.822
2 (50)	10*s*	0.109"	2.373	2.975	4.287	4.287	4.287	4.287
	40	0.154"	1.330	1.667	3.359	3.359	3.359	3.359
	80	0.218"	1.000	1.000	2.613	2.613	2.613	2.613
	160	0.344"	1.000	1.000	1.852	1.852	1.852	1.852
2½ (65)	10*s*	0.120"	2.089	2.619	3.614	4.580	4.580	4.580
	40	0.203"	1.000	1.085	1.497	3.161	3.161	3.161
	80	0.276"	1.000	1.000	1.000	2.528	2.528	2.528
	160	0.375"	1.000	1.000	1.000	2.008	2.008	2.008
3 (80)	10*s*	0.120"	2.161	2.710	3.739	4.624	5.249	5.249
	40	0.216"	1.000	1.012	1.397	1.728	3.480	3.480
	80	0.300"	1.000	1.000	1.000	1.000	2.747	2.747
	160	0.438"	1.000	1.000	1.000	1.000	2.073	2.073
4 (100)	10*s*	0.120"	2.257	2.829	3.904	4.828	5.583	6.239
	40	0.237"	1.000	1.000	1.250	1.546	1.788	3.893
	80	0.337"	1.000	1.000	1.000	1.000	1.000	3.030
	160	0.531"	1.000	1.000	1.000	1.000	1.000	2.168
6 (150)	10*s*	0.134"	2.005	2.513	3.469	4.289	4.960	6.250
	40	0.280"	1.000	1.000	1.012	1.251	1.447	1.823
	80	0.432"	1.000	1.000	1.000	1.000	1.000	1.000
	120	0.562"	1.000	1.000	1.000	1.000	1.000	1.000
8 (200)	10*s*	0.148"	1.776	2.227	3.073	3.800	4.394	5.537
	20	0.250"	1.000	1.000	1.280	1.583	1.830	2.306
	40	0.322"	1.000	1.000	1.000	1.037	1.199	1.510
	80	0.500"	1.000	1.000	1.000	1.000	1.000	1.000

1 in = 25.4 mm

4. *The requirement for the effects of any single nonrepeated anchor movement is not specified.*

ASME B31.3 Process Piping Code.[9] This code governs all piping within the property limits of facilities engaged in the processing or handling of chemical, petroleum, or related products. Examples are a chemical plant, petroleum refinery, loading terminal, natural gas processing plant, bulk plant, compounding plant, and tank farm. Excluded from the B31.3 code are piping carrying nonhazardous fluid with an internal gauge pressure less than 15 psi (103.5 kPa) and a temperature below 366°F (186°C); plumbing; sewers; fire protection systems; boiler external piping per B31.1 as well as pipelines per B31.4 or B31.8.

TABLE B4.4*b* SIFs for Class 3000 Half-Couplings (Branch Pipe Schedule 40) (Run Pipe Size: NPS 10 to 24 (DN 250 to 600))

Run pipe			Sch. 40 branch pipe size and thickness					
			Branch pipe size					
			½ (15)	¾ (20)	1 (25)	1¼ (32)	1½ (40)	2 (50)
			Branch pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.109"	0.113"	0.133"	0.140"	0.145"	0.154"
10 (250)	10*s*	0.165"	1.538	1.928	2.660	3.290	3.804	4.793
	20	0.250"	1.000	1.000	1.329	1.644	1.901	2.395
	40	0.365"	1.000	1.000	1.000	1.000	1.010	1.272
	80	0.594"	1.000	1.000	1.000	1.000	1.000	1.000
12 (300)	10*s*	0.180"	1.369	1.716	2.368	2.928	3.386	4.267
	20	0.250"	1.000	1.000	1.368	1.692	1.957	2.466
	STD	0.375"	1.000	1.000	1.000	1.000	1.000	1.252
	40	0.406"	1.000	1.000	1.000	1.000	1.000	1.097
14 (350)	10	0.250"	1.000	1.007	1.390	1.719	1.988	2.505
	20	0.312"	1.000	1.000	1.000	1.187	1.373	1.730
	STD	0.375"	1.000	1.000	1.000	1.000	1.010	1.273
	40	0.438"	1.000	1.000	1.000	1.000	1.000	1.000
16 (400)	10	0.250"	1.000	1.030	1.422	1.759	2.034	2.562
	20	0.312"	1.000	1.000	1.000	1.215	1.405	1.770
	STD	0.375"	1.000	1.000	1.000	1.000	1.033	1.302
	40	0.500"	1.000	1.000	1.000	1.000	1.000	1.000
18 (450)	10	0.250"	1.000	1.051	1.451	1.794	2.075	2.614
	20	0.312"	1.000	1.000	1.002	1.239	1.433	1.806
	STD	0.375"	1.000	1.000	1.000	1.000	1.054	1.328
	30	0.438"	1.000	1.000	1.000	1.000	1.000	1.025
20 (500)	10	0.250"	1.000	1.070	1.477	1.826	2.112	2.661
	STD	0.375"	1.000	1.000	1.000	1.000	1.073	1.352
	30	0.500"	1.000	1.000	1.000	1.000	1.000	1.000
	40	0.593"	1.000	1.000	1.000	1.000	1.000	1.000
24 (600)	10	0.250"	1.000	1.103	1.523	1.883	2.178	2.744
	STD	0.375"	1.000	1.000	1.000	1.000	1.107	1.395
	XS	0.500"	1.000	1.000	1.000	1.000	1.000	1.000
	30	0.562"	1.000	1.000	1.000	1.000	1.000	1.000

1 in = 25.4 mm

The loadings required to be considered are pressure, weight (live and dead loads), impact, wind, earthquake-induced horizontal forces, vibration, discharge reactions, thermal expansion and contraction, temperature gradients, and anchor movements.

The governing equations are as follows:

1. *Stresses due to sustained loads.* The sum of the longitudinal stresses S_L due to pressure, weight, and other sustained loads must not exceed S_h (basic allowable stress at maximum metal temperature). The thickness of pipe used in calculating S_L shall be the nominal thickness minus mechanical, corrosion, and crosion allowances.

TABLE B4.4c SIFs for Class 3000 Half-Couplings (Branch Pipe Schedule 80) (Run Pipe Size: NPS 1½ to 8 (DN 40 to 200))

Run pipe			Sch. 80 branch pipe size and thickness					
			Branch pipe size					
			½ (15)	¾ (20)	1 (25)	1¼ (32)	1½ (40)	2 (50)
			Branch pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.147"	0.154"	0.179"	0.191"	0.200"	0.218"
1½	10*s*	0.109"	2.841	3.665	3.665	3.665	3.665	3.665
(40)	40	0.145"	1.759	2.989	2.989	2.989	2.989	2.989
	80	0.200"	1.024	2.362	2.362	2.362	2.362	2.362
	160	0.281"	1.000	1.822	1.822	1.822	1.822	1.822
2	10*s*	0.109"	2.954	3.792	4.287	4.287	4.287	4.287
(50)	40	0.154"	1.655	2.124	3.359	3.359	3.359	3.359
	80	0.218"	1.000	1.184	2.613	2.613	2.613	2.613
	160	0.344"	1.000	1.000	1.852	1.852	1.852	1.852
2½	10*s*	0.120"	2.600	3.337	4.580	4.580	4.580	4.580
(65)	40	0.203"	1.077	1.382	1.898	3.161	3.161	3.161
	80	0.276"	1.000	1.000	1.132	2.528	2.528	2.528
	160	0.375"	1.000	1.000	1.000	2.008	2.008	2.008
3	10*s*	0.120"	2.690	3.453	4.742	5.249	5.249	5.249
(80)	40	0.216"	1.005	1.290	1.772	2.239	3.480	3.480
	80	0.300"	1.000	1.000	1.020	1.290	2.747	2.747
	160	0.438"	1.000	1.000	1.000	1.000	2.073	2.073
4	10*s*	0.120"	2.809	3.605	4.951	6.239	6.239	6.239
(100)	40	0.237"	1.000	1.154	1.585	2.004	2.351	3.893
	80	0.337"	1.000	1.000	1.000	1.110	1.302	3.030
	160	0.531"	1.000	1.000	1.000	1.000	1.000	2.168
6	10*s*	0.134"	2.496	3.203	4.398	5.560	6.523	7.534
(150)	40	0.280"	1.000	1.000	1.283	1.622	1.903	2.470
	80	0.432"	1.000	1.000	1.000	1.000	1.000	1.194
	120	0.562"	1.000	1.000	1.000	1.000	1.000	1.000
8	10*s*	0.148"	2.211	2.838	3.897	4.925	5.779	7.502
(200)	20	0.250"	1.000	1.182	1.623	2.052	2.407	3.125
	40	0.322"	1.000	1.000	1.063	1.344	1.576	2.046
	80	0.500"	1.000	1.000	1.000	1.000	1.000	1.000

1 in = 25.4 mm

2. *Stresses due to occasional loads.* The sum of the longitudinal stresses due to pressure, weight, and other sustained loads and of the stresses produced by occasional loads such as earthquake or wind shall not exceed $1.33S_h$. Earthquake and wind loads need not be considered as acting simultaneously.

3. *Stress range due to expansion loads.* The displacement stress range S_E shall not exceed S_A:

$$S_E \leq S_A \tag{B4.28}$$

where $S_E = (S_b^2 + 4S_t^2)^{1/2}$
S_b = resultant bending stress, psi
$= [(i_i M_i)^2 + (i_o M_o)^2]^{1/2}/Z$

TABLE B4.4*d* SIFs for Class 3000 Half-Couplings (Branch Pipe Schedule 80) (Run Pipe Size: NPS 10 to 24 (DN 250 to 600))

Run pipe			Sch. 80 branch pipe size and thickness					
			Branch pipe size					
			½ (15)	¾ (20)	1 (25)	1¼ (32)	1½ (40)	2 (50)
			Branch pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.147"	0.154"	0.179"	0.191"	0.200"	0.218"
10 (250)	10*s*	0.165"	1.914	2.457	3.373	4.264	5.003	6.494
	20	0.250"	1.000	1.227	1.685	2.130	2.499	3.245
	40	0.365"	1.000	1.000	1.000	1.132	1.328	1.724
	80	0.594"	1.000	1.000	1.000	1.000	1.000	1.000
12 (300)	10*s*	0.180"	1.704	2.187	3.003	3.796	4.453	5.781
	STD	0.375"	1.000	1.000	1.000	1.114	1.307	1.697
	40	0.406"	1.000	1.000	1.000	1.000	1.144	1.486
	80*s*	0.500"	1.000	1.000	1.000	1.000	1.000	1.049
14 (350)	10	0.250"	1.000	1.284	1.763	2.228	2.614	3.394
	STD	0.375"	1.000	1.000	1.000	1.132	1.328	1.724
	40	0.438"	1.000	1.000	1.000	1.000	1.024	1.330
	XS	0.500"	1.000	1.000	1.000	1.000	1.000	1.066
16 (400)	10	0.250"	1.023	1.313	1.803	2.279	2.674	3.472
	20	0.312"	1.000	1.000	1.246	1.575	1.847	2.398
	STD	0.375"	1.000	1.000	1.000	1.158	1.359	1.764
	40	0.500"	1.000	1.000	1.000	1.000	1.000	1.091
18 (450)	10	0.250"	1.044	1.340	1.840	2.325	2.728	3.542
	STD	0.375"	1.000	1.000	1.000	1.182	1.386	1.800
	30	0.438"	1.000	1.000	1.000	1.000	1.070	1.388
	40	0.562"	1.000	1.000	1.000	1.000	1.000	1.000
20 (500)	10	0.250"	1.062	1.364	1.873	2.367	2.777	3.605
	STD	0.375"	1.000	1.000	1.000	1.203	1.411	1.832
	30	0.500"	1.000	1.000	1.000	1.000	1.000	1.133
	40	0.593"	1.000	1.000	1.000	1.000	1.000	1.000
24 (600)	10	0.250"	1.096	1.406	1.931	2.441	2.864	3.718
	STD	0.375"	1.000	1.000	1.000	1.241	1.456	1.890
	XS	0.500"	1.000	1.000	1.000	1.000	1.000	1.169
	30	0.562"	1.000	1.000	1.000	1.000	1.000	1.000

1 in = 25.4 mm

M_i = in-plane bending moment, in · lb
M_o = out-of-plane bending moment, in · lb
i_i = in-plane stress intensification factor obtained from Fig. B4.5*a* (see also figure note 10)
i_o = out-of-plane stress intensification factor obtained from Fig. B4.5*a* (see also figure note 10)
S_t = torsional stress, psi
= $M_t/(2Z)$
M_t = torsional moment, in · lb

TABLE B4.4*e* SIFs for Class 6000 Half-Couplings (Branch Pipe Schedule 160) (Run Pipe Size: 1½ to 8 in (DN 40 to 200))

Run pipe			Sch. 160 branch pipe size and thickness					
			Branch pipe size					
			½ (15)	¾ (20)	1 (25)	1¼ (32)	1½ (40)	2 (50)
			Branch pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.188"	0.219"	0.250"	0.250"	0.281"	0.344"
1½	10*s*	0.109"	2.763	3.664	3.665	3.665	3.665	3.665
(65)	40	0.145"	1.711	2.459	2.989	2.989	2.989	2.989
	80	0.200"	1.000	1.431	2.362	2.362	2.362	2.362
	160	0.281"	1.000	1.822	1.822	1.822	1.822	1.822
2	10*s*	0.109"	2.873	4.128	4.287	4.287	4.287	4.287
(50)	40	0.154"	1.610	2.313	2.979	3.359	3.359	3.359
	80	0.218"	1.000	1.290	1.661	2.613	2.613	2.613
	160	0.344"	1.000	1.000	1.852	1.852	1.852	1.852
2½	10*s*	0.120"	2.529	3.633	4.580	4.580	4.580	4.580
(65)	40	0.203"	1.048	1.505	1.939	3.161	3.161	3.161
	80	0.276"	1.000	1.000	1.157	2.528	2.528	2.528
	160	0.375"	1.000	1.000	1.000	2.008	2.008	2.008
3	10*s*	0.120"	2.617	3.759	4.843	5.249	5.249	5.249
(80)	40	0.216"	1.000	1.405	1.809	2.481	2.859	3.480
	80	0.300"	1.000	1.000	1.042	1.429	2.747	2.747
	160	0.438"	1.000	1.000	1.000	1.000	2.073	2.073
4	10*s*	0.120"	2.732	3.925	5.056	6.239	6.239	6.239
(100)	40	0.237"	1.000	1.257	1.619	2.220	2.558	3.642
	80	0.337"	1.000	1.000	1.000	1.230	1.417	2.017
	160	0.531"	1.000	1.000	1.000	1.000	1.000	1.168
6	10*s*	0.134"	2.427	3.487	4.492	6.159	7.098	7.534
(150)	40	0.280"	1.000	1.017	1.310	1.796	2.070	2.947
	80	0.432"	1.000	1.000	1.000	1.000	1.001	1.425
	120	0.562"	1.000	1.000	1.000	1.000	1.000	1.000
8	10*s*	0.148"	2.150	3.089	3.980	5.456	6.288	8.425
(200)	20	0.250"	1.000	1.287	1.658	2.273	2.619	3.728
	40	0.322"	1.000	1.000	1.086	1.488	1.715	2.442
	80	0.500"	1.000	1.000	1.000	1.000	1.000	1.168

1 in = 25.4 mm

S_A = allowable displacement stress range
$= f(1.25S_c + 0.25S_h)$
$= f[1.25(S_c + S_h) - S_L]$ when $S_h > S_L$
S_c = basic allowable stress at minimum metal temperature, psi
f = stress range reduction factor per Table B4.2

ASME B31.4 Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia, and Alcohols Piping Code.[10] The scope of ASME B31.4, Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia, and Alcohols, governs piping transporting liquids such as crude oil,

TABLE B4.4f SIFs for Class 6000 Half-Couplings (Branch Pipe Schedule 160) (Run Pipe Size: 10 to 24 in (DN 250 to 600))

Run pipe			Sch. 160 branch pipe size and thickness					
			Branch pipe size					
			½ (15)	¾ (20)	1 (25)	1¼ (32)	1½ (40)	2 (50)
			Branch pipe wall thickness					
NPS (DN)	Sch.	Thickness	0.188"	0.219"	0.250"	0.250"	0.281"	0.344"
10 (250)	10*s*	0.165"	1.862	2.674	3.445	4.723	5.444	7.749
	20	0.250"	1.000	1.336	1.721	2.360	2.720	3.872
	40	0.365"	1.000	1.000	1.000	1.254	1.445	2.057
	80	0.594"	1.000	1.000	1.000	1.000	1.000	1.000
12 (300)	10*s*	0.180"	1.657	2.381	3.067	4.205	4.846	6.898
	STD	0.375"	1.000	1.000	1.000	1.234	1.422	2.024
	40	0.406"	1.000	1.000	1.000	1.081	1.245	1.773
	80	0.687"	1.000	1.000	1.000	1.000	1.000	1.000
14 (350)	10	0.250"	1.000	1.398	1.800	2.468	2.845	4.050
	STD	0.375"	1.000	1.000	1.000	1.254	1.445	2.057
	40	0.438"	1.000	1.000	1.000	1.000	1.115	1.587
	60	0.594"	1.000	1.000	1.000	1.000	1.000	1.000
16 (400)	10	0.250"	1.000	1.430	1.842	2.525	2.910	4.142
	STD	0.375"	1.000	1.000	1.000	1.283	1.479	2.105
	40	0.500"	1.000	1.000	1.000	1.000	1.000	1.301
	60	0.656"	1.000	1.000	1.000	1.000	1.000	1.000
18 (450)	10	0.250"	1.015	1.458	1.879	2.576	2.969	4.226
	STD	0.375"	1.000	1.000	1.000	1.309	1.508	2.147
	30	0.438"	1.000	1.000	1.000	1.010	1.164	1.657
	40	0.562"	1.000	1.000	1.000	1.000	1.000	1.092
20 (500)	10	0.250"	1.033	1.485	1.912	2.622	3.022	4.302
	STD	0.375"	1.000	1.000	1.000	1.333	1.536	2.186
	30	0.500"	1.000	1.000	1.000	1.000	1.000	1.352
	40	0.593"	1.000	1.000	1.000	1.000	1.000	1.014
24 (600)	10	0.250"	1.066	1.531	1.972	2.704	3.116	4.436
	STD	0.375"	1.000	1.000	1.002	1.374	1.584	2.255
	30	0.562"	1.000	1.000	1.000	1.000	1.000	1.147
	40	0.687"	1.000	1.000	1.000	1.000	1.000	1.000

1 in = 25.4 mm

condensate, natural gasoline, natural gas liquids, liquefied petroleum gas, liquid alcohol, liquid anhydrous ammonia, and liquid petroleum products between producers' lease facilities, tank farms, natural gas processing plants, refineries, stations, ammonia plants, terminals, and delivery and receiving points. Excluded from B31.4 are auxiliary piping such as water, air, steam, lubricating oil, gas, and fuel; piping with an internal gauge pressure at or below 15 psi (103.5 kPa) regardless of temperature; piping with an internal gauge pressure above 15 psi (103.5 kPa) and a temperature below -20°F (-29°C) or above 250°F (121°C); and piping for petroleum refinery, gas transmission and distribution, ammonia refrigeration, and so on, that is covered by other ASME B31 sections.

The limits of calculated stresses are as follows:

1. *Stresses due to sustained loads.* The sum of the longitudinal stresses due to pressure, weight, and other sustained external loads shall not exceed $0.75S_A$, where $S_A = 0.72S_y$ (specified minimum yield strength).
2. *Stresses due to occasional loads.* The sum of the longitudinal stresses produced by pressure, live and dead loads, and those produced by occasional loads, such as wind or earthquake, shall not exceed $0.8S_y$.
3. *Stresses due to expansion loads*

   a. Restrained lines. The net longitudinal compressive stress due to the combined effects of temperature rise and fluid pressure shall be computed from the equation.

$$S_L = E\alpha(T_2 - T_1) - \nu S_H \tag{B4.29}$$

where S_L = longitudinal compressive stress, psi
S_H = hoop stress due to fluid pressure, psi
T_1 = temperature at time of installation, °F
T_2 = maximum or minimum operating temperature, °F
E = modulus of elasticity, psi
α = linear coefficient of thermal expansion, in/(in · °F)
ν = Poisson's ratio = 0.30 for steel

Then the equivalent tensile stress is calculated as

$$S_{\mathrm{eqiv}} = S_H + S_L < 0.9S_y \tag{B4.30}$$

where S_{eqiv} = the equivalent tensile stress, psi. Beam bending stresses shall be included in the longitudinal stress for those portions of the restrained line which are supported aboveground.

   b. Unrestrained lines. Stresses due to expansion for those portions of the piping without substantial axial restraint shall be combined in accordance with the following equation:

$$S_E = \left(S_b^2 + 4S_t^2\right)^{1/2} < S_A \tag{B4.31}$$

where S_E = stress due to expansion, psi
S_b = $[(i_iM_i)^2 + (i_oM_o^2)]^{1/2}/Z$
= equivalent bending stress, psi
S_t = $M_t/(2Z)$ = torsional stress, psi
M_i = in-plane bending moment, in · lb
M_o = out-of-plane bending moment, in · lb
M_t = torsional moment, in · lb
i_i = in-plane stress intensification factor obtained from Fig. B4.5*a*
i_o = out-of-plane stress intensification factor obtained from Fig. B4.5*a*
Z = section modulus of pipe, in^3

ASME B31.5 Refrigeration Piping Code.[11] The scope of this code covers refrigerant and secondary coolant piping for temperatures as low as -320°F (196°C). Excluded from this code are piping designed for external or internal gauge pressure not exceeding

15 psi (103.5 kPa) regardless of size; water piping; and any self-contained or unit systems subject to the requirements of Underwriters' Laboratories or other nationally recognized testing laboratory.

The limits of calculated stresses are as follows:

1. *Stresses due to expansion loads.* The expansion stress range S_E shall not exceed the allowable stress range S_A:

$$S_E < S_A = f(1.25S_c + 0.25S_h) \tag{B4.32}$$

where S_E = expansion stress range = $(S_b^2 + 4S_t^2)^{1/2}$, psi
S_b = resultant bending stress
= $[(i_iM_i)^2 + (i_\sigma M_o)^2]^{1/2}/Z$, psi
S_t = torsional stress = $M_t/(2Z)$, psi
Mi = in-plane bending moment, in · lb
M_o = out-of-plane bending moment, in · lb
M_t = torsional moment, in · lb
i_i = in-plane stress intensification factor obtained from Fig. B4.5a
i_o = out-of-plane stress intensification factor obtained from Fig. B4.5a
Z = section modulus of pipe, in^3
S_c = basic material allowable stress at minimum (cold) normal temperature , psi
S_h = basic material allowable stress at maximum (hot) normal temperature , psi
f = stress range reduction factor obtained from Fig. B4.5*d*

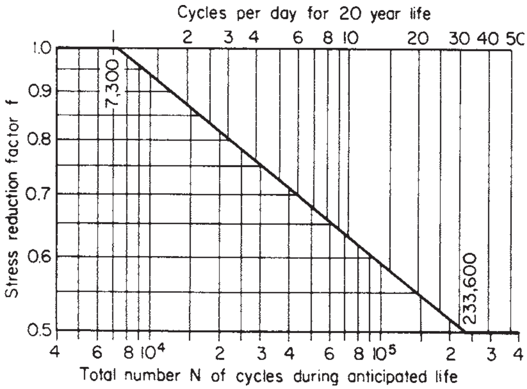


FIGURE B4.5*d* Stress range reduction factors. (*Extracted from Refrigeration Piping Code, ASME B 31.5 1992. Courtesy of ASME.*)

2. *Stresses due to sustained loads.* The sum of the longitudinal stresses due to pressure, weight, and other sustained external loading S_L shall not exceed S_h. Where $S_L > S_h$, the difference S_h - S_L may be added to the term in parentheses in Eq. (B4.32).

3. *Stresses due to occasional loads.* The sum of the longitudinal stresses produced by pressure, live and dead loads, and occasional loads, such as wind or earthquake, may not exceed $1.33S_h$. It is not necessary to consider wind and earthquake as occurring concurrently.

TABLE B4.5 Temperature Derating Factor T for Steel Pipe

Temperature, °F (°C)		Temperature derating factor T
250 or less	(121.1 or less)	1.000
300	(148.9)	0.967
350	(176.7)	0.933
400	(204.4)	0.900
450	(233.2)	0.867

Note: For intermediate temperatures, interpolate for derating factor.

Source: ASME B31.8, 1995. *Gas Transmission and Distribution Piping Systems. (Courtesy of ASME.)*

ASME B31.8, Gas Transmission and Distribution Piping Code.[12] This code governs most of the pipelines in gas transmission and distribution systems up to the outlet of the customer's meter set assembly. Excluded from this code are piping with metal temperatures above 450°F (232.2°C) or below -20°F (-29°C); piping beyond the outlet of the customer's meter set assembly; piping in oil refineries or natural gas extraction plants, gas treating plants, and so on, which is covered by other ASME B31 codes; waste gas vent pipe operating at atmospheric pressures; and liquid petroleum transportation piping. The governing equations are as follows:

1. *Stresses due to pressure and external loads.* The sum of the longitudinal pressure stress and the longitudinal bending stress due to external loads such as weight, wind, and so on, S_L, shall not exceed $0.75 S_y FT$:

$$S_L \leq 0.75 S_y F T \tag{B4.33}$$

where S_y = specified minimum yield strength, psi
T = temperature derating factor obtained from Table B4.5
F = construction-type design factor obtained from Table B4.6. The construction types are associated with the population density of the surrounding area as follows:
Type A: Sparsely populated areas such as deserts, mountains, and farmland
Type B: Fringe areas around cities or towns

TABLE B4.6 Values of Design Factor F

Construction type	Design factor F
A	0.72
B	0.60
C	0.50
D	0.40

Source: ASME B31.8, 1995. (*Courtesy of ASME.*)

Type C: Cities or towns with no buildings over three stories tall
Type D: Areas with taller buildings

2. *Stress range due to expansion loads.* The maximum combined expansion stress range S_E shall not exceed $0.72S_y$:

$$S_E = \left(S_b^2 + 4S_t^2\right)^{1/2} < 0.72S_y \tag{B4.34}$$

where S_b = resultant bending stress = iM_b/Z, psi
S_t = torsional stress = $M_t/(2Z)$, psi
M_b = resultant bending moment, in · lb
M_t = torsional moment, in · lb
i = stress intensification factor obtained from Fig. B4.5a (see figure note 10)
Z = section modulus of pipe, in^3

3. *Stresses due to pressure, external loads, and expansion loads.* The sum of the longitudinal pressure stress, the longitudinal bending stress due to external loads, and the combined stress due to expansion shall not exceed S_y.

STRESS INTENSIFICATION FACTORS (SIF) FOR NONSTANDARD FITTINGS

Stress intensification factors for fittings such as Wye (Y) connections and latrolets are not available in the current editions of codes. The following subsections provide data for obtaining appropriate SIFs for various nonstandard fittings.

SIF for 90-Degree Wye Connection

The following SIF (i) which can be used is based on a comparative finite element analysis study[13] of a Wye connection and a Tee connection. See Fig. B4.5*i*.

i for forged Wye connection = 1.0 (i of WTEE)

for all moments except for torsion and out-of-plane bending moment components from branch side

i_{ob} for forged Wye connection = 3.3 (i of WTEE)

for out-of-plane bending moment components from branch side

i_{tb} for forged Wye connection = 1.1 (i of WTEE)

for torsional moment components from branch side

Use UTEE (unreinforced tee) instead of WTEE (welding tee) in the above expressions if the Wye connection is a fabricated Wye.[13]

SIF for Weldolets, Sockolets, and Half-Couplings

$$i_r = 0.8\left(\frac{R}{T}\right)^{2/3}\frac{r}{R} \geq 2.1 \quad \text{(for run side)}$$

$$i_b = 1.5\left(\frac{R}{T}\right)^{2/3}\left(\frac{r}{R}\right)^{1/2}\frac{t}{T}\frac{r}{r_p} \geq 1.5 \quad \text{if } \frac{r}{R} \leq 0.9 \quad \text{(for branch side)}$$

$$i_b = 0.9\left(\frac{R}{T}\right)^{2/3}\frac{t}{T}\frac{r}{r_p} \geq 1.0 \quad \text{if } \frac{r}{R} = 1.0 \quad \text{(for branch side)}$$

For branch-side stresses, branch-side section modulus should be used.

For insert or contour weldolets, the coefficients in the above equations 0.8, 1.5, and 0.9 are replaced by 0.4, 0.75, and 0.45, respectively.

R, T = mean radius and wall thickness of run pipe
r, t = mean radius and wall thickness of branch pipe
r_p = outside radius of the reinforcement on nozzle or branch

The above equations are based on Refs. 14 and 15.

SIF for 45-Degree Latrolets

Forged latrolet: $i = 0.5727\ (R/T)^{2/3}$ based on $h = 1.97\ (T/R)$

Fabricated lateral: $i = 1.5378(R/T)^{2/3}$ obtained by multiplying the SIF of forged latrolet by $(4.4)^{2/3}$

See Ref. 16.

SIF for Reducers

Even though the reducer is a standard fitting and the formula for SIF is available in the codes,[17] the cone angle is not readily available. The following SIF formula from the codes can be used for large-bore concentric and eccentric reducers. Using data from Ref. 18, the following expressions can be used for the cone angle, α:

$$i = 0.5 + 0.01\alpha\left(\frac{d_2}{t_2}\right)^{0.5} \qquad (1.0 < i \leq 2.0)$$

$$\alpha_1 = \left(80.4\,\frac{D}{d}\right) - 62.9 \quad \text{for} \quad 1 < \frac{D}{d} < 1.3$$

$$\alpha_1 = \left(28.3\,\frac{D}{d}\right) + 2.9 \quad \text{for} \quad \frac{D}{d} \geq 1.3$$

$$\alpha = \frac{\alpha_1}{2} \quad \text{for concentric reducers}$$

$$\alpha = \alpha_1 \quad \text{for eccentric reducers}$$

where α = cone angle in degrees
D, d = mean diameter of large and small ends of reducer, respectively
d_2, t_2 = outside diameter and thickness of small end of reducer, respectively

See Refs. 17 and 18.

LOCAL STRESSES

In addition to the general pipe stresses (the pressure stress and the moment stress) as described in the previous sections, there are certain local pipe wall stresses produced by (1) restraint of the pipe radial thermal and internal pressure expansion of pipe-through-structural-steel type of anchors, (2) the transfer of load from the supporting surface to the pipe surface over a contact length along the axis of the pipe, or (3) attachments welded to pipe (e.g., lugs and trunnions).

Local Stresses and Code Requirements

The local stresses S_L, S_L^*, S_L^{**}, and S_L^{***} can be expressed as follows:

$$S_L,\, S_L^*,\, S_L^{**},\ \text{or}\ S_L^{***} = \max\left[|\sigma_1 + \sigma_1'|,\, |\sigma_2 + \sigma_2'|,\, |\sigma_2 + \sigma_2' - (\sigma_1 + \sigma_1')|\right] \quad \text{(B4.35)}$$

where S_L = local stress due to deadweight, psi
S_L^* = local stress due to deadweight, seismic inertia, and other dynamic loads, psi
S_L^{**} = local stress due to thermal expansion and seismic anchor movement, psi
S_L^{***} = local stress due to concurrently acting loads, psi
σ_1 = longitudinal membrane stress, psi
σ_2 = circumferential membrane stress, psi
σ_2' = circumferential bending stress, psi
σ_1' = longitudinal bending stress, psi

Strictly speaking, the present piping codes give no specific limits for local stresses. As an industry practice, the calculated local stress is added to the general pipe stress and then compared with the pipe stress allowables specified by the applicable code. As an example, the total (general plus local) pipe stresses for ASME Class 2 and 3 piping shall satisfy the following equations [see Eqs. (B4.9), (B4.10), (B4.11), (B4.12) for definitions of symbols]:

1. *Design loading*

$$\frac{B_1 P D_o}{2t} + \frac{B_2 M_A}{Z} + \frac{2}{3} S_L \leq 1.5 S_h \quad \text{(B4.36)}$$

2. *Service loadings*

$$\frac{B_1 P_{\max} D_o}{2t} + \frac{B_2(M_A + M_B)}{Z} + \frac{2}{3} S_L^*$$

$$\leq \begin{cases} 1.85 S_h, \text{ but not greater than } 1.5 S_y \text{ for level A and B loadings} \\ 2.25 S_h, \text{ but not greater than } 1.8 S_y \text{ for level C loadings} \\ 3.0 S_h, \text{ but not greater than } 2.0 S_y \text{ for level D loadings} \end{cases} \tag{B4.37}$$

3. *Sustained and thermal expansion loading*: Either of the following equations shall be satisfied:

$$\frac{iM_C}{Z} + \frac{1}{2} S_L^{**} \leq S_A \tag{B4.38}$$

or

$$\frac{PD_o}{4t} + \frac{0.75 i M_A}{Z} + \frac{iM_C}{Z} + \frac{2}{3} S_L + \frac{1}{2} S_L^{**} \leq S_h + S_A \tag{B4.39}$$

4. *Local stress limit loading*

$$S_L^{***} < 2.5 S_y \tag{B4.40}$$

Local Stress Due to Restraint of Pipe Radial Expansion

The membrane and flexural stresses can be calculated as follows[19]:

$$\sigma_1 \text{ is negligible}$$

$$\sigma_1' = \frac{\pm 12 D\lambda^2(\delta_{\widehat{P}} + \delta_T)}{t^2} \tag{B4.41}$$

$$\sigma_2 = \frac{-(\delta_P + \delta_T)E}{R} + \nu\sigma_1 \tag{B4.42}$$

$$\sigma_2' = \nu\sigma_1' \tag{B4.43}$$

$$\delta_P = \frac{PR^2}{Et} \tag{B4.44}$$

$$\delta_T = \alpha R \Delta T \tag{B4.45}$$

where P = internal pressure of pipe, psi
R = pipe outside radius, in
E = modulus of elasticity of pipe, psi
t = pipe wall thickness, in
α = coefficient of thermal expansion of pipe, in/(in · °F)
ΔT = range of thermal expansion temperatures, °F
ν = Poisson's ratio

$D = Et^3/[12(1 - v^2)]$
$\lambda = [3(1 - v^2)/(Rt)^2]^{1/4}$

In the case of the local stress produced by restraint to the pipe radial expansion, $S_L = S_L^* = 0$. For fillet weld, i = 2.1 should be used. In addition, the stress check on limit loading is required. Here, $S_L^{***} = S_L^{**}$.

Local Stress Due to Contact

The membrane and flexural stresses can be calculated as follows[19]:

1. *For line contact*

$$\sigma_1 = -0.52F(R_m)^{1/4}L^{-1/2}t^{-7/4} \tag{B4.46}$$

(*Note*: σ_1' is considered to be included in this equation.)

$$\sigma_2 = -0.496F(R_m)^{3/4}L^{-3/2}t^{-5/4} \tag{B4.47}$$

$$\sigma_2' = \pm 3.15\sigma_1 \tag{B4.48}$$

where F = support load, lb
R_m = mean radius of pipe, in
L = contact length, in
t = pipe wall thickness, in

2. *For point contact.* σ_1 and σ_1' are negligible compared to $\sigma_2 + \sigma_2'$:

$$\sigma_2 = \frac{0.4F}{t^2} \tag{B4.49}$$

$$\sigma_2' = \frac{2.4F}{t^2} \tag{B4.50}$$

3. In the case of contact stress, the minimum nominal general pipe stress (i.e., the unintensified general pipe stress) may be used in Eqs. (B4.36) to (B4.39) to calculate the total pipe stress. In addition, the stress check on limit loading is not required.

Other Types of Local Stresses

The two types of local stresses previously described are commonly encountered by stress analysts. Detailed descriptions and analysis methods for other types of local stresses such as the local stresses at integral welded attachments to pipe (e.g., lugs and trunnions) can be found in technical publications, *Welding Research Council Bulletins 107 and 198*, and ASME Code Cases.[21-23]

ANALYSIS OF INTEGRAL WELDED ATTACHMENTS (IWA)

Integral Welded Attachments are often used to support piping systems. The local stresses in the piping at IWA locations are commonly evaluated using the Welding

Research Council (WRC) Bulletin #107 approach,[21] which is based on Bijlaard's[20] work. Generally, the various methods for local stress evaluations can be categorized in accordance with the following list. (Friction-induced loads due to weight and thermal expansion, if applicable, should be included.)

1. Stress intensification factor (SIF) approach for certain configurations
2. WRC Bulletin #107 approach with limitation on ß (attachment parameter) and γ (shell parameter) parameters
3. ASME Code cases approach
4. Approach based on utilization of any available finite element analysis (FEA) results or published data
5. Rigorous FEA approach

SIF Approach

In this method, the local stresses are not evaluated directly but are indirectly accounted for by applying a SIF to the general piping stresses.

The SIF approach has the following limitations:

- Applicable to some specific IWA configurations only
- Not applicable to lugs, irregular shapes, and so forth

The SIF approach is applicable in the following situations:

- *360° (full) wrapper plates*. This configuration is no longer a local stress problem. A SIF (i) of 2.1 or 1.3 can be applied, depending on the applicable code.
- *180° wrapper plates*. The following SIFs are recommended by Rodabaugh (see Fig. B4.5e):

  i = 4.2 for the run pipe torsional moment (M_{tr}) component

  i = 2.1 for the run pipe out-of-plane bending moment (M_{obr}) component

  i = 1.3 for the run pipe in-plane bending moment (M_{ibr}) component
- *Circular trunnion/stanchions on straight pipe*. Consider the configuration as a reinforced tee (RTEE) and intensify the general piping stresses using a SIF (i) of

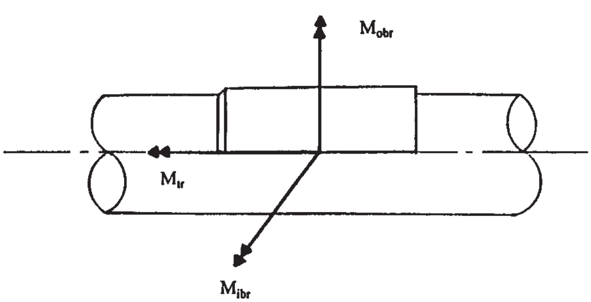


FIGURE B4.5*e* 180° wrapper plate (IWA).

RTEE per the applicable code requirements. Since there is no hole made in the pipe's pressure boundary, the run pipe thickness can be considered as a reinforcement. If there is a pad, the pad thickness can be considered as an additional reinforcement. Of course, the codes limit the effective thickness of reinforcement [$(t_e)_{max} = 1.5 \times t$].

- *Attachments on fittings.* Cross multiplication of SIFs (for example, a round attachment on elbow or on a tee) can be used (elbow SIF × RTEE SIF or TEE SIF × RTEE SIF).

WRC Bulletin #107 Approach

This is based on the analytical work performed by Professor P.P. Bijlaard of Cornell University and subsequently generated experimental data. The results are presented in WRC Bulletin #107 (WRC-107) as nondimensional curves based on β and γ parameters for three different types of loading. The local stresses can be evaluated by hand by filling out the computation sheets for local stresses as given in the WRC-107 bulletin or utilization of commercially available computer program software which has stored the nondimensional curves as digitized data. The WRC-107 method generally yields conservative results. The following additional information relates to this approach:

- *Basis.* Bijlaard's approach is based on shell theory and some simplifications for radial load (P), longitudinal moment (M_L), and circumferential moment (M_C).
- *Strength-of-materials formulas.* Simple strength-of-materials formulas are used to compute shear stresses due to longitudinal shear load (V_L), circumferential shear load (V_C), and torsional moment (M_T) loadings.
- *Shell and attachment parameters and loadings.* The shell, attachment parameters, and loadings are as follows (see Figs. B4.5*f* and B4.5*g*):

  shell parameter $\gamma = D_m/(2T)$ where $D_m = D_o - T$

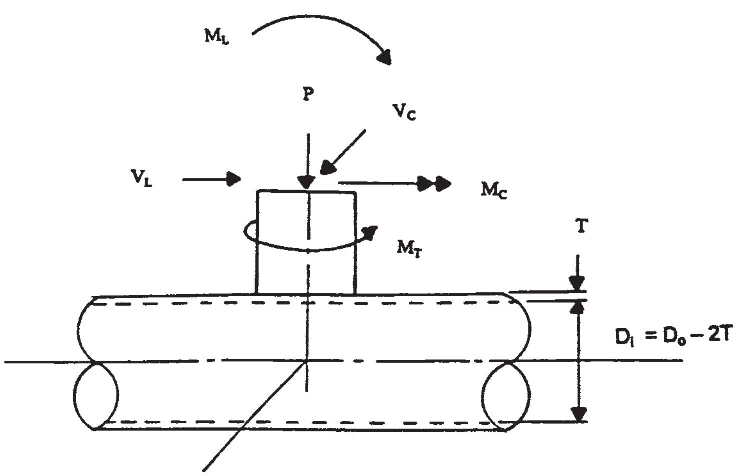


FIGURE B4.5*f* IWA notations.

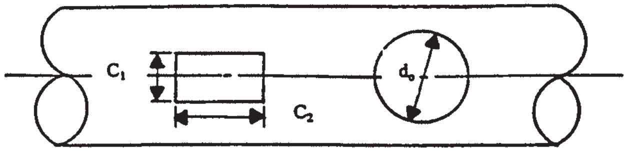


FIGURE B4.5g IWA notations (*continued*).

attachment parameter (β):
For circular attachment $\beta = 0.875\ (d_o/D_m)$
For square or rectangular attachment

$$\beta_1 = C_1/D_m$$

$$\beta_2 = C_2/D_m$$

Caution: WRC-107 uses C_1 and C_2 to represent one-half of the attachment dimensions in circumferential and longitudinal directions, while here C_1 and C_2 are used for the full attachment dimensions.

- *Limitations.* The WRC Bulletin #107 approach has the following limitations:

$$0.01 \le \beta \le 0.5$$
$$5 \le \gamma \le 300$$
$$1/4 \le \beta_1/\beta_2 \le 4 \text{ for } M_c$$
$$1/4 \le \beta_2/\beta_1 \le 4 \text{ for } M_L$$

- *Shear stresses.* The formulas for shear stress calculations are as follows:

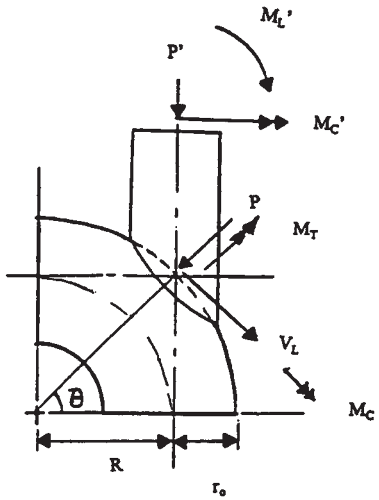
	Square/rectangular IWA	Circular IWA
V_L	$\tau = V_L/(2C_2T)$	$\tau = 2V_L/(\pi d_o T)$
V_C	$\tau = V_C/(2C_1T)$	$\tau = 2V_C/(\pi d_o T)$
M_T	$\tau = M_T/F^*$ (see Ref. 22)	$\tau = 4M_T/[2\pi(d_o)^2T]$

*

$$F = \text{larger of} \begin{cases} T\{[C_{\max} + C_{\min}](C_{\min}/2)\}; \text{ or} \\ \{[1.57 + 0.093(C_{\max}/C_{\min})](C_{\min})^2(C_{\max})/8\} \end{cases}$$

where $C_{\max}$ = maximum of C_1 and C_2
$C_{\min}$ = minimum of C_1 and C_2

- *Attachments on elbows.* To evaluate local stresses in elbows with attachments, the following approach can be utilized. Attachment loads (P', M'_L, M'_C) can be resolved at the elbow/attachment interface to components P, V_L, M_c, M_T, M_L. An equivalent



$P = P' \sin\theta$; $V_L = P' \cos\theta$

$M_L = M'_L$

$M_C = M_C' \sin\theta$; $M_T = M_C' \cos\theta$

Where; $\cos\theta = R/(R + r_o)$; R = radius of Elbow;
r_o = Outside radius of Pipe

FIGURE B4.5*b* IWA on elbow.

straight pipe with the attachment and resolved loads can be considered in local stress evaluation using WRC-107 (see Fig. B4.5*b*).

Approach Based on ASME Code Cases

The Welding Research Council (WRC) Bulletin #198[22] fitted equations with some inherent conservatism to curves of WRC Bulletin #107. The results are published as ASME Code cases. Interested readers can refer to the ASME Code cases listed in the following table[23] for details and limitations of their applicability. Generally speaking, the local stress results from Code cases are more conservative than WRC Bulletin #107 results.

	ASME Code Class 1	ASME Code Classes 2 and 3
Rectangular attachments	Code Case # N-122 (1745)	Code Case # N-318
Hollow circular attachments	Code Case # N-391	Code Case # N-392

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Approach Based on Available FEA Results

For some commonly utilized configurations and sizes, FEA was performed and the results were compared with WRC-107 results. Reduction factors are supplied relative to WRC-107 results. See Ref. 24.

Based on extensive finite element analysis studies of certain sizes, shapes, and configurations of attachments, the factors in the following tables are generated. The P, M_L, M_c loads can be reduced by dividing them with the applicable factors, and then the WRC-107 approach can be used. The γ value should be limited to the following range: $3.5 \leq \gamma \leq 31.5$.

Factors for Attachments on Straight Pipe

		P		M_L		M_C	
		σ_m	$\sigma_m + \sigma_b$	σ_m	$\sigma_m + \sigma_b$	σ_m	$\sigma_m + \sigma_b$
Small attachments	S	1.23	1.28	1.11	1.36	1.00	1.21
$\beta \leq 0.5$	C	1.56	2.25	1.03	1.42	1.00	1.46
	R	1.17	1.46	1.08	1.36	1.00	1.32
	R (Note 1)	1.00	1.46	1.00	1.36	0.42	1.32
Large attachments	R	1.80	1.50	1.25	2.02	1.15	1.89
$0.5 < \beta \leq 0.7$	C	1.30	1.99	1.00	1.35	0.74	1.47
Note 2							
Large attachments	R	1.81	2.58	1.94	3.53	1.70	2.41
$\beta > 0.7$	C	2.29	3.33	1.80	2.57	1.23	2.22
Note 2							
Large attachments	R	3.6	4.0	4.61	4.86	7.52	19.04
$\beta > 0.7$ and extending 180° circumferentially	C	1.07	2.12	7.77	14.57	3.87	15.14
Note 2							

Notes: 1. Factors for rectangular attachments when $3.5 \leq \gamma < 5$ and $\beta \leq 0.5$.
2. Apply these factors to the loads when actual $\beta > 0.5$ and evaluate the local stress based on WRC-107 utilizing artificially reduced attachment size to correspond to $\beta = 0.5$.

Notation: *S:* square attachments
R: rectangular attachments
C: circular attachments
σ_m: membrane stress
σ_b: bending stress
P: radial load
M_L: longitudinal moment
M_C: circumferential moment

See ASME paper listed in Ref. 24.

Factors for Circular Attachments on Long-Radius Elbows

	P		M_L		M_C	
	σ_m	$\sigma_m + \sigma_b$	σ_m	$\sigma_m + \sigma_b$	σ_m	$\sigma_m + \sigma_b$
$0.5 < d_o/D_o \leq 0.8$	2.00	3.60	1.53	2.66	1.61	2.87
$0.8 < d_o/D_o \leq 1$	3.14	4.88	2.42	3.53	2.52	4.37

Notes: Apply these factors to the loads (divide the loads by these factors) and perform WRC-107 evaluation using attachment size reduced to correspond to $ß = 0.5$.

Notation: d_o: outside diameter of circular attachment

D_o: outside diameter of pipe

See ASME paper listed in Ref. 24.

Rigorous FEA Approach

For this approach, a 3-D finite-element model of the shell and attachment has to be built. A portion of the pipe, attachment, and pipe/attachment interface can be modeled and loadings applied. FEA analysis can be performed and the local stresses can be evaluated.

Although cumbersome and time-consuming, FEA is becoming a viable option due to the availability of commercial FEA software which can run on personal computer platforms. Personal computers have become very powerful tools.

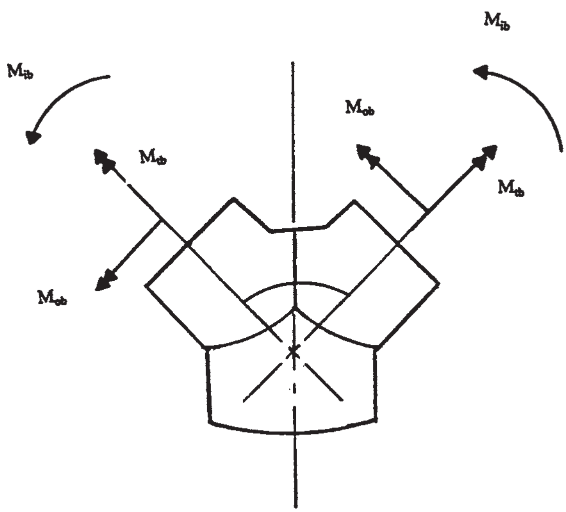


FIGURE B4.5*i* Wye connection.

TYPES OF PIPE LOADING CONDITIONS

1. *Types of loads.* As previously mentioned in the subsection "Classification of Loads," piping loads are classified into three types: *sustained loads, occasional loads,* and *expansion loads.* These three types of loads and the corresponding analysis will be discussed in this section in detail.
2. *Method of analysis.* The piping stress analysis to be performed could be a simplified analysis or a computerized analysis. The choice of the proper analysis depends on the pipe size and the piping code. For small (nominal diameter 2 in and under) pipe except nuclear Class 1 pipe, a cookbook-type, simplified analysis could be performed. For nuclear Class 1 piping, since the requirements are more stringent, a computerized analysis is required. A detailed description of a cookbook-type, simplified analysis and a brief description of a computerized analysis are given in the section that follows, "Methods of Analysis." Generally, before computerized analysis is performed, pipe supports may be located using the cookbook method.

Sustained Load: Pressure

Internal pressure in piping usually induces stresses in the pipe wall rather than loads on the pipe supports. This is because pressure forces are balanced by tension in the pipe wall, resulting in zero pipe support loadings. A discussion of unbalanced forces in the pipe created by pressure waves during fluid transients is given in the subsection "Dynamic Loads."

Pressure Stress. The longitudinal stress developed in the pipe due to internal pressure can be calculated as follows:

$$S_{LP} = \frac{PD}{4t}$$

or

$$S_{LP} = \frac{Pd^2}{D^2 - d^2} = P\left(\frac{A_f}{A_m}\right) \tag{B4.51}$$

where S_{LP} = longitudinal stress, psi
P = internal design pressure, psig
D = outside diameter, in
d = inside diameter, in
A_f = flow area, in^2
A_m = metal area, in^2
t = pipe wall thickness, in

The second equation gives pressure stress in terms of the ratio of pipe flow area to metal area. It also provides a more accurate result. Both equations are acceptable to the code.

Expansion Joint. In piping design, elbows, bends, and pipe expansion loops normally provide adequate flexibility for piping thermal expansion and contraction. However, in some cases this flexibility may not be adequate. As a solution, expansion joints may be used to absorb the expansion and contraction of pipe.

In general, expansion joints are used for the following applications:

1. Where thermal movements would induce excessive stress in normal piping arrangements
2. Where space is inadequate
3. Where reactions transmitted by pipe supports or anchors create large loads on supporting structures
4. Where reactions to equipment terminals are in excess of allowables

When expansion joints are used in piping, the pressure forces can no longer be balanced by tension in the pipe wall, and the pressure forces will be resisted by pipe supports and anchors.

There are many types of expansion joints available, ranging from a piece of rubber hose to metal bellows. The metal bellows expansion joint is most commonly used for power or process piping. Figure B4.6 shows the various components of a bellows expansion joint.

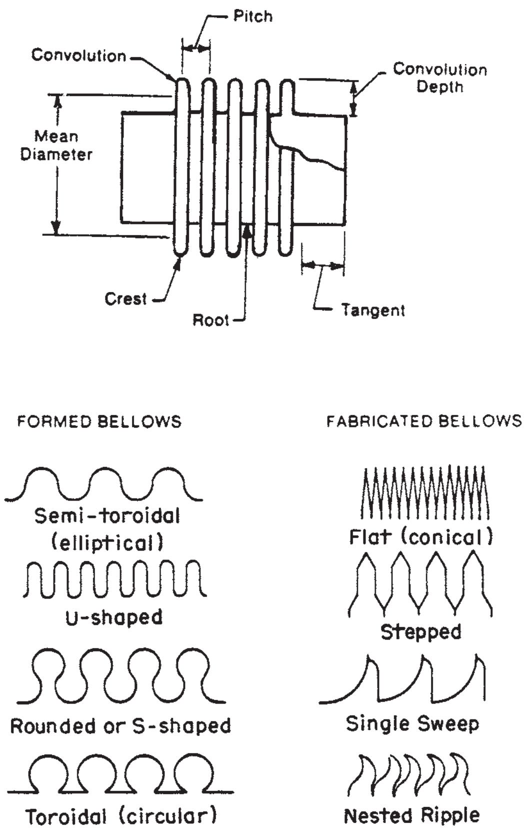


FIGURE B4.6 Bellows expansion joints.

Expansion joints do not have the capability to transmit large pressure forces. Restraints are usually installed on both sides of the expansion joint to prevent the pressure force from pulling the joint apart. The pressure force developed in the expansion joint is equal to the internal pressure times the maximum cross-sectional area over which it is applied. Since an expansion joint increases the flexibility of a piping system, the flexibility (spring rate) of the expansion joint should be incorporated in the piping stress analysis. Typical axial spring rates of bellows can be found in Ref. 25.

Sustained Load: Weight

The total design weight load of pipe supports includes the weight of the pipe, fittings, insulation, fluid in pipe, piping components such as valves, valve operators, flanges, and so on, and the supports themselves. Supports should be located as specified in Chap. B5.

Hydrotest and Other Occasional Loadings. To assure the integrity and leak tightness of a piping system designed to Section III of ASME Boiler and Pressure Vessel Code or ASME B31.1, the codes require that a pressure test be performed prior to placing the system in service. The most commonly used test is the hydrostatic test. When a steam or gas piping system is to be hydrotested, the effects of the weight of the water on the system and its supports must be considered. A hydroweight stress analysis should be performed to assure that the pipe supports, which have been designed for the normal operating condition, are able to withstand the hydrotest loads. If permanent supports cannot withstand these hydrotest loads, temporary supports may be added. Spring supports are available with hydrostatic test stops, which, in effect, transform the units into rigid supports.

Whether or not required by code, other conditions, such as the added weight of a cleaning medium of density greater than that of the process fluid, must be considered in a manner similar to that discussed above. Both dynamic and static loading analyses may be impacted by flushing and blowing-out activities during construction or after major rework.

Thermal Expansion Loads

For weight analysis, the more pipe supports installed, the lower the stress developed in the pipe. However, the opposite is true for the case of piping thermal expansion. When thermal expansion of the piping due to fluid or environmental temperature is restrained at supports, anchors, equipment nozzles, and penetrations, large thermal stresses and loads are caused.

Thermal Modes. Piping systems are generally analyzed for one thermal condition or mode, that is, the maximum operating temperature. However, piping systems that have more than one operating mode with different operating temperatures concurrently in different parts of the piping system should be analyzed for these operating thermal modes.

With the aid of system flow diagrams or *piping and instrumentation drawings (P&ID)*, the stress analyst can determine the thermal modes required for a particular piping system. For B31.1 piping and ASME Class 2 and 3 piping, the required thermal modes can be determined by using good engineering judgment in selecting the most severe thermal conditions. For ASME Class 1 piping, the required thermal modes can be determined by examining the load histograms specified in the design specification.

Free Thermal Analysis. During the initial stage of piping analysis, an unrestrained (i.e., no intermediate pipe supports) or free thermal analysis may be performed. This' analysis is performed for the worst thermal mode and includes only terminal points such as penetrations, anchors, and equipment nozzles. The result of this free thermal analysis usually gives useful information, which can be utilized by the stress analyst in the later stages of the piping analysis. Generally, a resulting thermal expansion stress < 10 ksi (68,948 kPa) means adequate flexibility exists in the piping system. The piping locations with low resulting thermal displacements would be good locations where rigid supports may be installed without adversely affecting the flexibility of the piping system. The resulting equipment nozzle loads could be used to evaluate the capabilities of the equipment for meeting the equipment manufacturer's nozzle allowables.

Imposed Thermal Movements. Thermal expansion of equipment causes displacements in the attached piping. Thermal stresses may also be caused due to thermal anchor movements at terminal ends and intermediate restraints. Therefore, appropriate thermal analysis for thermal anchor movements relating to the respective thermal modes should also be performed. Sometimes, it is possible for thermal anchor movements to exist when the piping is cold. In such cases, analysis in the cold condition, with only the thermal anchor movements as input, may be required.

LOCA Thermal Analysis. In nuclear power plants, following a *loss-of-coolant accident (LOCA)*, the containment (the building structure designed to contain fission products) expands due to the rise in temperature and pressure inside the containment. This containment thermal growth results in large containment penetration anchor movements which affect the connected piping. It is not required to qualify the piping for this faulted condition. Thermal analysis for these LOCA anchor movements is used only for the evaluation of flanges, equipment nozzle loads, and pipe support loads.

Temperature Decay. For piping systems having a portion of the system with stagnant branch lines (dead legs) as shown in Fig. B4.7, it is necessary to consider the temperature

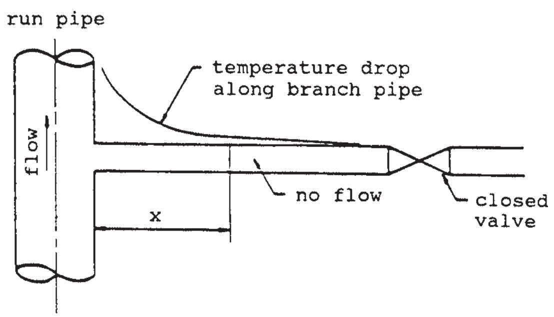


FIGURE B4.7 Temperature decay at dead leg.

decay in the piping. One simple approach to this temperature attenuation problem is as follows:

1. For a piping system with water, the temperature of the branch pipe is assumed to be the same as the run pipe up to a length equivalent to 10 times the inside pipe diameter. The remaining portion of the branch pipe may be considered at ambient temperature.
2. For a piping system with steam or gas, the temperature of the branch pipe is considered the same as the run pipe up to the closed valve.

For cases such as thermal transient analysis of ASME Class 1 piping, where a more accurate temperature profile along the branch pipe may be required, the approach described in Ref. 26 should be used.

Stress Ranges. The thermal stresses developed in the pipe are in fact "stress ranges," that is, the difference between the unit thermal expansion for the highest operating temperature and for the lowest operating temperature.

For piping systems that do not experience temperatures below ambient temperature, the stress range is the difference between the unit expansion for the maximum thermal mode and that for 70°F (21°C). (See later subsection "Seismic Anchor Movement and Building Settlement Analysis.")

For systems with supply from a pool or river which might go below 70°F (21°C) in the winter, negative coefficients of expansion should be considered in evaluating the stress range.

Occasional Loads: Seismic

The code of Federal Regulation 10CFR Part 50 requires that safety-related piping in nuclear power plants be designed to withstand seismic loadings without loss of capability to perform their function.[27] For nonnuclear piping in regions of high seismic activity, this design requirement should also be considered.

OBE and SSE. Nuclear piping systems and components classified as Seismic Category I are designed to withstand two levels of site-dependent hypothetical earthquakes: the *safe-shutdown earthquake (SSE)* and the *operational-basis earthquake (OBE)*.[28]

For conservatism, the OBE must usually be equal to at least one-half of the SSE. Their magnitudes are expressed in terms of the gravitational acceleration g. Their motions are assumed to occur in three orthogonal directions: one vertical and two horizontal.

Seismic Category I systems are defined as those necessary to assure:

1. The integrity of the reactor coolant pressure boundary
2. The capability to shut down the reactor and maintain it in a safe shutdown condition
3. The capability to prevent or mitigate potential off-site radiation exposure

Types of Seismic Analysis. Generally, piping seismic analysis is performed through one of three methods: *time-history analysis, modal response spectrum analysis,* or *static analysis*.

The equation of motion for a piping system subjected to an externally applied loading (seismic excitation) may be expressed as

$$M\ddot{\mathbf{x}} + C\dot{\mathbf{x}} + K\mathbf{x} = \mathbf{f} \tag{B4.52}$$

where M = mass matrix of system
C = damping matrix
K = stiffness matrix
$\ddot{\mathbf{x}}$ = acceleration vector
$\dot{\mathbf{x}}$ = velocity vector
$\mathbf{x}$ = displacement vector
$\mathbf{f}$ = external loading vector, function of time

This equation could be solved by time-history analysis.

Time-History Analysis. Time-history analysis is based on hypothetical earthquake data in the form of ground displacement, velocity, or acceleration versus time. The piping system is represented by lumped masses connected by massless elastic members. The analysis is performed on this mathematical model by the direct numerical integration method.[29,30] At each time step, the piping stresses, displacements, and restraint loads are calculated. Time history simulates the behavior of the piping system during the seismic excitation. The main advantage of time-history analysis is that analytically it is more accurate and less conservative compared to other approaches. The main disadvantages of time-history analysis are the excessive computational time required and the difficulty of obtaining a realistic earthquake input time function.

Modal Response Spectrum Analysis. The seismic response spectrum is a plot of the maximum acceleration response of a number of idealized single-degree-of-freedom oscillators attached to the floor (structure) with certain damping.

These response spectra are based on design response spectra and specified maximum ground accelerations of the plant site. Usually, a series of curves with different damping values for operating and design basis earthquakes for each orthogonal direction are generated, as shown in Fig. B4.8.

In the modal response spectrum analysis, the piping system is idealized as lumped masses connected by massless elastic members. The lumped masses are carefully located to adequately represent the dynamic properties of the piping system.

After the stiffness and mass matrix of the mathematical model are calculated, the natural frequencies of the piping system and corresponding mode shapes for all significant modes of vibration are also determined using the following equation:

$$\left(K - W_n^2 M\right)\phi_n = 0 \tag{B4.53}$$

where K = stiffness matrix
W_n = natural circular frequency for the nth mode
M = mass matrix
ϕ_n = mode shape matrix for the nth mode

The modal spectral acceleration taken from the appropriate response spectrum is then used to find the maximum response of each mode:

$$(Y_n)_{\max} = \frac{\phi_n^t M D S a_n}{W_n^2 M_n} \tag{B4.54}$$

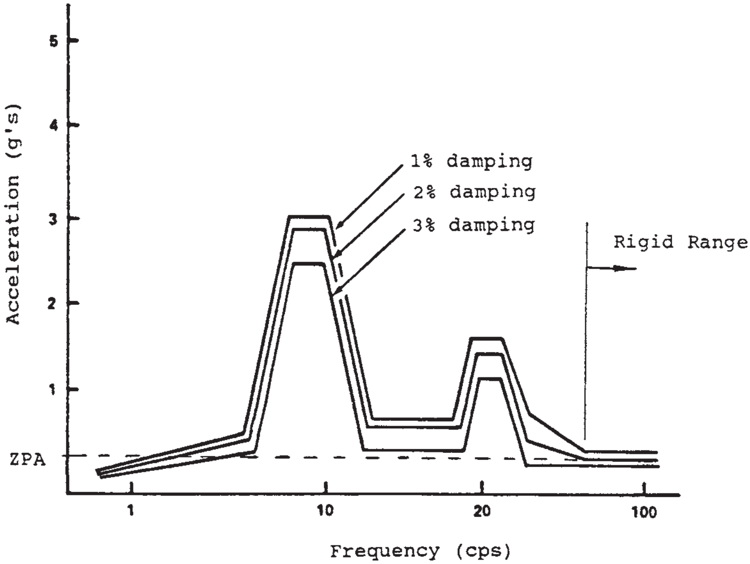


FIGURE B4.8 Response spectrum curves.

where Sa_n = spectral acceleration value for the nth mode
D = earthquake direction coefficient
ϕ_n^t = transpose of the nth mode shape
M_n = generalized mass of the nth mode
Y_n = generalized coordinate for the nth mode

Using the maximum generalized coordinate for each mode, the maximum displacements, the effective inertia forces, the effective acceleration, and the internal forces and moments associated with each mode are calculated as follows:

$$X_n = \phi_n (Y_n)_{\max}$$
$$F_n = K X_n$$
$$a_n = M^{-1} F_n$$
$$L_n = b F_n \qquad \text{(B4.55)}$$

where X_n = displacement matrix due to nth mode
F_n = effective inertia force matrix due to nth mode
a_n = effective acceleration matrix due to nth mode
M^{-1} = the inverse of mass matrix
L_n = internal force and moment matrix due to nth mode
b = force transformation matrix

These modal components are then combined by the appropriate method (see later subsection "Methods for Combining System Responses") to obtain the total displacements, accelerations, forces, and moments for each point in the piping system.

Two types of response spectrum analyses can be performed depending on the pipe routing and attachments to buildings and structures.

Single-Response Spectrum Analysis. This type of analysis is performed using an enveloped response spectrum curve that covers all buildings and elevations to which the piping system is attached.

Multiple-Response Spectrum Analysis. This type of analysis is used where the piping is attached to various buildings or structures that have a wide variation in the amplitude or frequency of accelerations. In such cases, various response spectra curves may be applied at corresponding support and anchor points in the piping system.[31,32]

Static Analysis. Static analysis may be used to evaluate power piping or some piping systems in nuclear power plants. It is performed by analyzing a piping system for the statically applied uniform load equivalent to the site-dependent earthquake accelerations in each of the three orthogonal directions. All rigid restraints and snubbers supporting the pipe in the direction of the earthquake acceleration are included in the analysis. The total seismic effect is obtained by combining the results of the three directions.

The minimum earthquake force for structures described in ANSI A58.1[33] is also one form of static seismic analysis. The code recommends that a lateral seismic force will be assumed to act nonconcurrently in the direction of each of the main axes of the structure in accordance with the formula:

$$V = ZIKCSW \tag{B4.56}$$

where V = lateral seismic force, lb

Z = numerical coefficient, dependent upon the earthquake zone (see Fig. B4.9), 0.1 for Zone 0, 0.25 for Zone 1, 0.50 for Zone 2, and 1.00 for Zone 3

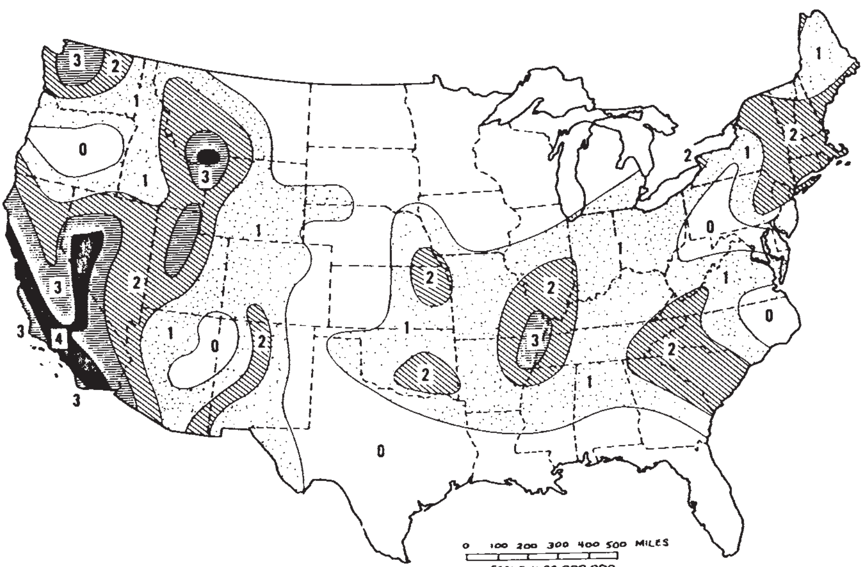


FIGURE B4.9 Map for seismic zones, contiguous 48 states. (*ANSI A58.1, 1982. Courtesy of ANSI.*)

TABLE B4.7 Dampling Values (Percent of Critical Damping)

Structure or component	Operating-basis earthquake or ½ safe-shutdown earthquake	Safe-shutdown earthquake
Equipment and large-diameter piping systems, pipe diameter greater than 12 in	2	3
Small-diameter piping systems, diameter equal to or less than 12 in	1	2
Welded steel structures	2	4
Bolted steel structures	4	7
Prestressed concrete structures	2	5
Reinforced concrete structures	4	7

Source: U.S. Nuclear Regulatory Commission. Regulatory Guide 1.61.

I = occupancy importance factor, usually between 1.0 and 1.5
K = horizontal force factor, dependent upon the arrangement of lateral force-resisting elements, usually between 0.67 and 2.50
C = $1/(15T^{1/2})$ but not to exceed 0.12
T = fundamental period of structure, s
S = soil factor, dependent upon the soil profile type, usually between 1.0 and 1.5
W = total dead weight of structure, lb

Damping. Damping is the phenomenon of dissipation of energy in a vibrating system. Each damping value expressed as a percentage of the critical damping is represented in the seismic response spectrum by a separate curve. The higher the damping value, the lower would be the effects of the seismic excitation. The damping values to be used for different levels of the earthquake are given by the NRC (U.S. Nuclear Regulatory Commission) Regulatory Guide 1.61,[34] as shown in Table B4.7.

When a system has both categories of pipe sizes mentioned in the table, dual damping values should be considered in the analysis.

Alternative damping values for response spectrum analysis of ASME Classes 1, 2, and 3 piping are given in ASME Code Case N-411-1,[35,36] as shown in Fig. B4.10. These damping values are applicable to both OBE and SSE. They are also independent of pipe size. As can be seen from Fig. B4.10, the damping values of Code Case N-411-1 are generally higher than the damping values given in Regulatory Guide 1.61. The industry has been applying these higher damping values to existing piping systems to reduce the number of snubbers installed in the plants in order to save snubber maintenance cost. The use of Code Case N-411-1 is acceptable to the NRC subject to the conditions described in the NRC Regulatory Guide 1.84.[37]

Mass Point Spacing. In a seismic analysis, the piping is represented by lumped masses connected by massless elastic members. The locations of these lumped masses are referred to as the *mass points*. In order to accurately represent the piping, the mass points on straight runs of pipe should be no farther apart than a length of pipe which would have a fundamental frequency of 33 Hz (see the later subsection "Cookbook-Type Analysis"). Mass points should also be located at all supports, concentrated weights such as valves, valve operators, flanges, and strainers, and at the end of

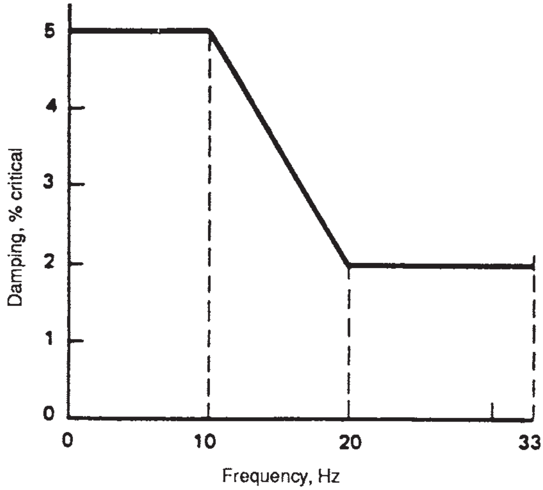


FIGURE B4.10 Code Case N-411 damping values. (*ASME B & PV Code, Case N-411-1, February 1989. Courtesy of ASME.*)

cantilevered vents and drains. At least two mass points should be placed between supports in the same direction.[38]

Cutoff Frequency, Rigid Range, Zero Period Acceleration, and Missing-Mass Effect. Generally, the piping response spectrum analysis is terminated at a frequency called the *cutoff frequency.* The cutoff frequency is usually specified as the frequency beyond which the spectral acceleration remains constant, and this constant spectral acceleration is known as the *zero period acceleration (ZPA)* (see Fig. B4.8).

Supposing a piping system is so designed and supported that the first mode is higher than the cutoff frequency; then as far as the computer program is concerned, this piping system does not receive seismic excitation at all. Consequently, the result of this seismic analysis is invalid because of the artificial constraint specified by the stress analyst.

This phenomenon, known as the *missing-mass effect,*[39,40] *could also occur in the* following cases:

1. On pipe runs with axial restraint (support, anchor, or nozzle) where the longitudinal frequency could be higher than the cutoff frequency

2. Concentrated masses in a piping system supported in such a manner that the frequency of that portion of piping is high

Most of the computer programs normally used for piping stress analysis have the capability to evaluate the missing-mass effect. These programs usually utilize the acceleration from the spectrum at the cutoff frequency (ZPA) to calculate the missing-mass effect.

Methods for Combining System Responses. In general, there are two approaches for the combination of system responses. One approach, the *absolute sum method (ABS),* adds the peak system responses. The second approach, *square root-sum-of-squares method (SRSS),* gives a combined response equal to the square root of the sum of the squares of the peak responses. The SRSS method is preferred over the ABS method because not all the peak responses occur simultaneously.

In a response spectrum modal analysis, if the modes are not closely spaced (two consecutive modes are defined as closely spaced if their frequencies differ from each other by less than 10 percent of the lower frequency), responses could be combined by taking the SRSS method. For closely spaced modes, the NRC suggests that the method of combining the responses by the SRSS method may not be conservative. An acceptable method of grouping the closely spaced modes of vibration and combining the responses is described in the NRC Regulatory Guide 1.92.[41,42]

Seismic Anchor Movement and Building Settlement Analysis

A piping system, supported from two seismically independent structures that move out of phase during a seismic event, will experience stresses due to the differential displacement of the supports.

Buried pipe could be considered as supported by the soil. A differential movement during a seismic event between the soil and the building to which the pipe is routed could also cause stresses in the pipe.

Similarly, the differential settlements between two structures or between a building and the adjacent soil will induce stresses in piping which is routed between them.

Seismic Anchor Movement (SAM) Analysis. A seismic anchor movement analysis is required on a piping system where:

1. The piping is supported from two seismically independent structures, or
2. The piping is attached to large equipment having its own modes of vibration (e.g., steam generator, pressurizer, reactor vessel, or reactor coolant pump).

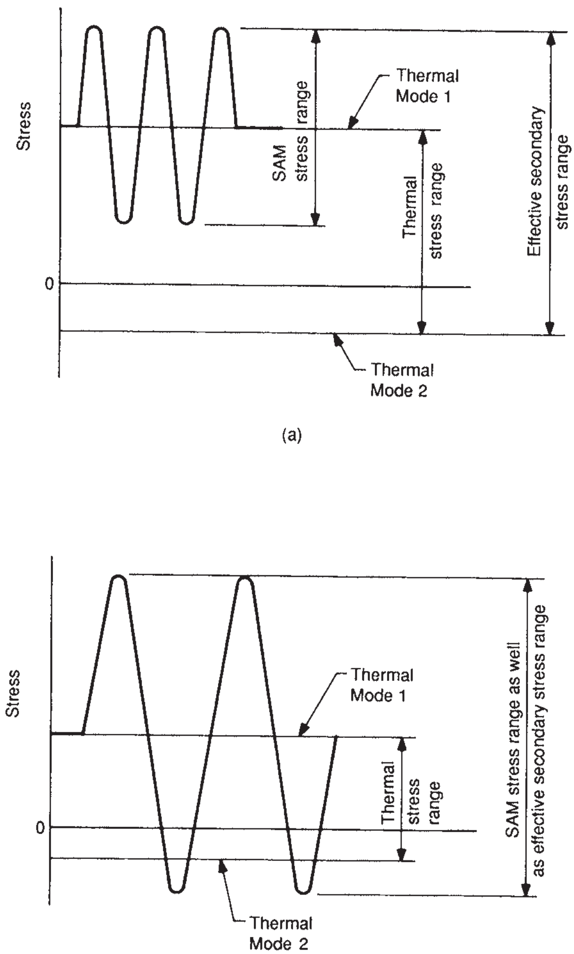
SAM analysis is performed by applying the corresponding seismic displacements of the building and structures at the pipe support and anchor locations. It is usually analyzed by a static method. However, dynamic supports such as snubbers and rigids (including anchors and nozzles) will be active while spring supports remain passive.

SAM displacements from the same building or structure are generally in phase, while those from different buildings or structures are considered out of phase.

When a terminal end of a piping system being analyzed is at a large pipe, the seismic movements from the large pipe analysis should be applied as a SAM displacement in the analysis.

The code allows the consideration of the stress due to SAM as either primary stress [see Eq. (B4.10)] or secondary stress [see Eq. (B4.11)]. However, it will usually be evaluated as secondary stress. Since the stress due to SAM is a cyclic type of stress, it should be combined with other cyclic-type secondary stresses such as thermal expansion stresses.

The total secondary stress range should include the thermal and SAM stress range. If the SAM stress is less than the thermal stress range, the effective secondary stress range is the sum of the SAM stress and the thermal stress range, as shown in Fig.



(b)

FIGURE B4.11 Effective secondary stress range.

B4.11*a*. If the SAM stress is higher than the thermal stress range, the effective secondary stress range then equals twice the SAM stress, as shown in Fig. B4.11*b*.

Building Settlement Analysis. ASME Boiler and Pressure Vessel Code Section III requires that the stresses due to building settlement be evaluated and be considered as secondary stresses. However, the stress due to building settlement is a one-time (single nonrepeated) anchor movement. Therefore, it is not required to combine it with other stresses. From Subsection NC-3653.2(b) of the code, the effects of any single nonrepeated anchor movement shall meet Eq. (B4.13).

Dynamic Loads

The dynamic loads discussed herein are limited to occasional loads (other than seismic loads) frequently encountered in piping stress analysis.

Safety Relief Valve Discharge Analysis. Safety-relief valves are installed for the purpose of protecting the fluid system from accidental overpressure, or venting the fluid generated in excess of requirement.

The general requirements pertaining to the design of the pressure relief discharge piping are provided in Appendix II of ASME B31.1, Subsections NC-3677 and NB-3677 of the ASME Code for different pipe classes.

There are two types of pressure relief valve discharge, namely, *open discharge* and *closed discharge,* as shown in the figures of Chap. B3.

Open Discharge. A typical open discharge is the transient due to discharging of steam from a steam line to the atmosphere through relief valves or safety valves. When the steam line pressure reaches the valve set point, the valve opens and decompression waves will travel both upstream and downstream of the valve. This flow transient sets up pressure imbalances along each pipe segment (a straight run of pipe bounded by elbows). The transient forces can be calculated by a computerized method as described in the later subsection "Steam Hammer-Water Hammer Analysis," while the reaction force at the valve exit due to steady-state flow is determined relatively easily.

Closed Discharge. In a closed-discharge system, the fluid is transmitted to its terminal receiver through continuous discharge piping. A typical closed discharge is the transient induced by a sudden opening of the relief and safety valves located on top of the pressurizer in a power plant. A water seal, which is maintained upstream of each valve to minimize leakage, driven by this high discharge pressure, generates a transient thrust force at each pipe segment. The water seal is discharged ahead of the steam as the valve disk lifts. For discharge piping with a water seal, only the first cycle of each event has a transient force based on water in the seal. The remaining cycles would be based on steam occupying the seal piping, and the transient forces would be reduced in magnitude.

Static Analysis. The static method of open discharge described in Appendix II of ASME B31.1 can be summarized as follows:

1. The reaction force F due to steady-state flow following the opening of the valve may be computed by

$$F = \frac{WV}{g} + (P - P_a)A \tag{B4.57}$$

where F = reaction force at exit, lbf
W = mass flow rate, lbm/s
V = exit velocity, ft/s
g = gravitational constant
= 32.2 lbm · ft/lbf · s^2
P = static pressure at exit, psia
P_a = atmospheric pressure, psia
A = exit area, in^2

2. The dynamic load factor (DLF) is used to account for the increased load caused by the sudden application of the discharge load. The DLF value will range between

1.1 and 2.0, depending on the time history of the applied load and the natural frequency of the piping.[43] If the run pipe is rigidly supported and the applied load could be assumed to be a single ramp function, the DLF may be determined in the following manner:

a. Calculate the safety valve installation period T:

$$T = 0.1846\left(\frac{Wh^3}{EI}\right)^{1/2} \tag{B4.58}$$

where T = safety valve installation period, s
W = weight of safety valve, installation piping, flanges, attachments, etc., lb
h = distance from run pipe to centerline of outlet piping, in
E = Young's modulus of inlet pipe, psi, at design temperature
I = moment of inertia of inlet pipe, in^4

b. Calculate the ratio t_o/T where to is the time the safety valve takes to go from fully closed to fully open (seconds).

c. For the ratio t_o/T, determine the DLF from data given in Appendix II of ASME B31.1, as shown in Fig. B4.12.

3. The moment due to valve reaction force is calculated by simply multiplying the force times the distance from the point in the piping system being analyzed, times a suitable DLF. The stress is then calculated accordingly.

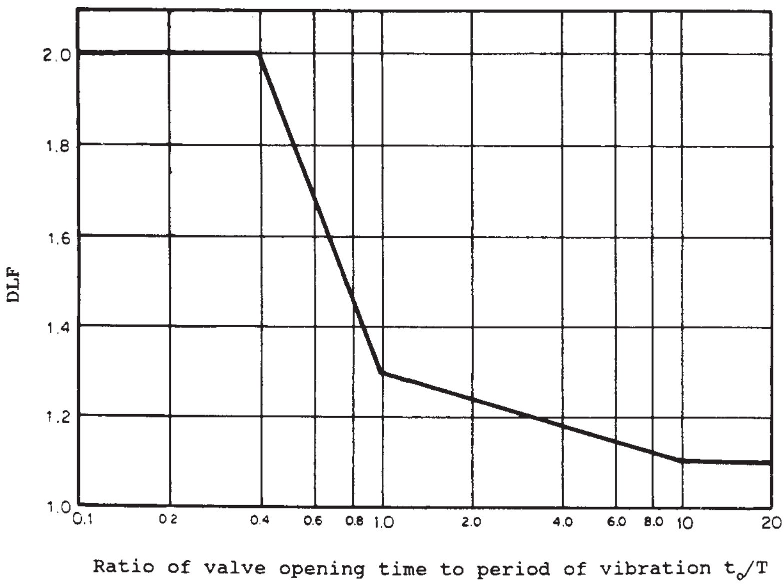


FIGURE B4.12 Hypothetical dynamic load factor (DLF).

Dynamic Analysis. The reaction force effects are dynamic in nature. A time-history dynamic analysis of the discharge piping is considered to be more accurate. Furthermore, closed-discharge systems do not easily lend themselves to simplified analysis techniques. A time-history analysis (such as the one described in the following subsection) is required to achieve realistic results.

Steam Hammer-Water Hammer Analysis. The steam hammer-water hammer event is often initiated by intentional actuation of certain flow control devices (main steam stop valve closure, feedwater pump trip, etc.), but in other cases a transient event could be introduced unintentionally as the result of some unforeseen operating condition, component malfunction, or accident (e.g., feedwater line check valve slam following a pipe break upstream of the check valve).[44,45] While these events may produce very complex transient fluid flow, the pipe stress analyst is interested in just the unbalanced force along the pipe segment tending to induce piping vibration.

Calculation of Unbalanced Forces. These time-history unbalanced forces are usually generated through a two-step computerized calculation. The fluid system is modeled as an assemblage of control volumes (e.g., piping volumes or steam generator) interconnected by junctions (e.g., valves, pump, or break). Piping fluid flow data, such as flow area, friction losses, valve closing-opening time, feed pump characteristics, or break characteristics, together with fluid initial conditions (flow rate, pressure, temperature, and mixture quality) are supplied as input to a thermal hydraulic finite difference computer program.[46]

Using this input information and a built-in steam table (fluid thermodynamic state), the first step solves the three equations of conservation (mass, momentum, and energy) at each time step for fluid properties such as pressure, velocity, internal energy, and mixture quality. A typical stop valve closure time history and its associated dynamic pressure time history are shown in Fig. B4.13. The second step utilizes a postprocessor. This postprocessor then accepts the output information from the first step and computes the unbalanced forces in piping segments by applying the momentum theorem.

Static Analysis. Static analysis is simple and saves computer time. It is used when the unbalanced forces are small and the total transient time is long. In the analysis, the peak values of the time-history fluid forcing functions at pipe segments are applied statically to the piping. The piping stress, deflections, and support-nozzle loads are then calculated by the computer program.

To obtain a conservative result for the static analysis, care must be taken in applying a proper dynamic load factor to the unbalanced forces.

Dynamic Analysis. The dynamic analysis generally utilizes either the direct step-by-step integration method (as described in the subsection "Pipe Break Analysis") or the modal-superposition method. In the dynamic analysis, the piping system is idealized as a mathematical model consisting of lumped masses connected by weightless elastic members. These lumped masses are carefully located to adequately represent the dynamic characteristics of the piping system. For computer programs utilizing the modal-superposition method, enough modes (or appropriate cutoff frequency) should be specified in the computer input such that the dynamic response of the piping system can be truly represented. There are no specific guidelines to damping values used in piping fluid transient dynamic analysis in the ASME Code or NRC published material. Therefore, it is recommended to use the OBE damping values prescribed in the NRC Regulatory Guide 1.61. Alternative damping values of Code Case N-411-1 are not applicable to the dynamic analysis.

The time-history unbalanced forces are applied to all pipe segments. Snubbers and rigid supports are effective restraints for transient forces. However, axial supports

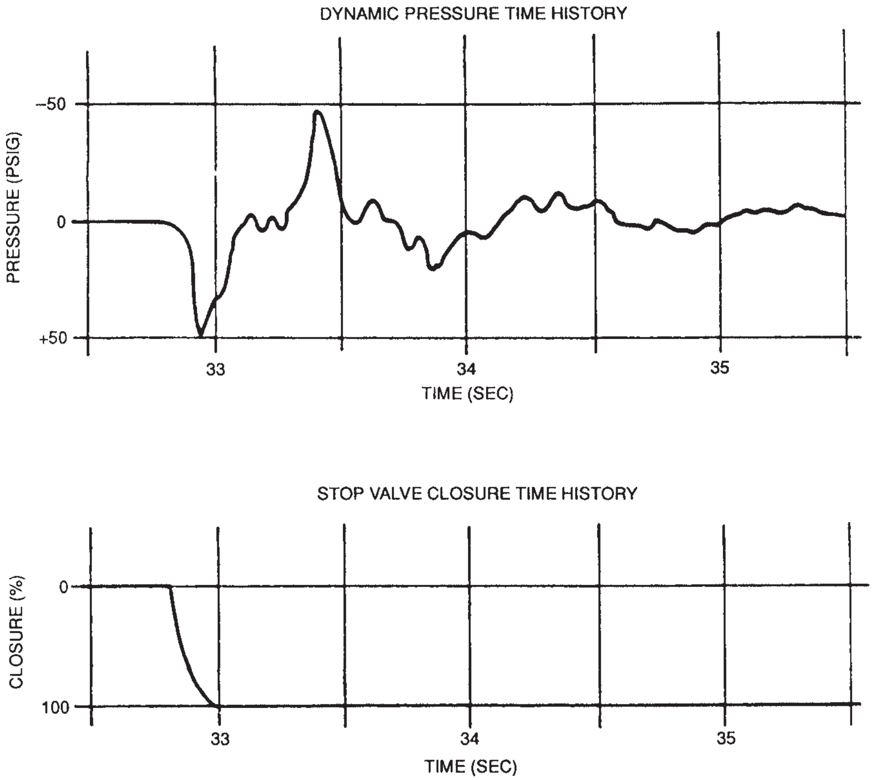


FIGURE B4.13 Steam hammer flow transients.

should be avoided in general. An axial support not only requires welded attachments on the pipe but also a pair of supports, which doubles the cost. To support the pipe axially, lateral supports can be used around the elbows. In addition, snubbers should not be located in the immediate vicinity of equipment nozzles. Snubbers located in such areas may not be activated during a fluid transient because of the dead band (built-in manufacturing tolerance) of the snubber hardware.

Stress Allowables. For the steam hammer–water hammer (e.g., feed pump trip) event, the pipe stress from the analysis is combined with stresses due to pressure, deadweight, and OBE in meeting the upset stress allowable. For piping in the turbine building, OBE stress is not included in the stress combination. For some water hammer (e.g., check valve slam) events, the stress from the analysis is combined with stresses due to pressure and deadweight in meeting the faulted stress allowable.

LOCA Analysis. LOCA (loss-of-coolant accident) is a postulated accident that results from the loss of reactor coolant, at a rate in excess of the capability of the reactor coolant makeup system, from breaks in the reactor coolant pressure boundary. Analyses should be performed by the *nuclear steam supply system (NSSS)* vendor to confirm the structural design adequacy of the reactor internals and reactor coolant piping (unbroken loop) to withstand the loadings of the most severe LOCA in combination with SSE per the requirements of 10CFR Part 50, Appendix A,[20] and the NRC Standard Review Plan 3.9.2.[47]

The integrity of the secondary system piping (main steam, feedwater, blowdown lines) off the steam generators also has to be assured by the architect-engineer (AE). Additional analyses to demonstrate the structural adequacy of some of the branch piping attached to the broken loop may be required by the NSSS vendor.[48] The information provided herein is limited to the secondary system piping off the steam generators.

Static Analysis. If a substantial separation between the forcing frequencies of the LOCA loading and the natural frequencies of the piping system can be demonstrated, a static analysis may be performed. In the static analysis, the maxima of each of the LOCA displacement components (three deflections and three rotations) are separately applied to the junctions of the *reactor coolant loop (RCL)* and the secondary system piping. The results should be combined absolutely and multiplied by an appropriate dynamic load factor.

Dynamic Analysis. The dynamic analysis can be performed in one of the following two ways:

Time-History Analysis. The LOCA displacement time history is applied dynamically to the junctions of the RCL and the secondary system piping. The damping value prescribed in the NRC Regulatory Guide 1.61 for SSE is suggested for this dynamic LOCA analysis. The detailed analysis method is similar to that described in the following subsection "Pipe Break Analysis."[49]

Response Spectrum Analysis. Compared to the time-history analysis, the response spectrum method is favorable for its low computer cost. However, this method may be unnecessarily conservative since the same loading has to be applied to the entire piping system. Because of the nature of the LOCA break and the impacting of the gapped RCL supports, the LOCA motion has much higher frequency content than the seismic excitation. The ZPA of a typical LOCA motion spectrum for a RCL junction is usually higher than that of a typical SSE response spectrum. Therefore, a higher cutoff frequency should be used in the analysis.[50]

Stress Allowables. The resulting stress from the LOCA analysis for the secondary piping system is combined with the stresses due to pressure, dead weight, and SSE in meeting the faulted stress allowables.

Pipe Break Analysis. Although it is extremely improbable that a pipe break will occur as postulated, public safety and the NRC licensing requirements make it necessary that such events must be considered in the design of high-energy piping systems.

A *high-energy piping system* is a piping system that, during normal plant conditions, is maintained at a temperature > 200°F (93.3°C), or a pressure > 275 psig (1896.1 kPa).

Pipe Break Locations. Pipe breaks are postulated in high-energy piping based on the primary plus secondary stresses and the cumulative usage factor.

1. *ASME Section III, Class 1 Pipe:* Pipe breaks are postulated to occur at terminal ends (the extremities of piping connected to structures, components, or anchors) and at all intermediate locations where:

   a. The primary plus secondary stress intensity range, as calculated by Equation (10) of Subsection NB-3653 [i.e., Eq. (B4.4) of this chapter], exceeds $2.4S_m$ and either Equation (12) or (13) [i.e., Eq. (B4.5) of this chapter] exceeds $2.4S_m$.

   b. Cumulative usage factor exceeds 0.1.

2. *ASME Section III, Class 2 and 3 Pipe:* Pipe breaks are postulated to occur at terminal ends and at all intermediate locations where the primary plus secondary stresses, as calculated by the sum of Equations (9) and (10) of Subsection NC-3653 [i.e., Eqs. (B4.10) and (B4.11) of this chapter], exceed $0.8(1.2S_h + S_A)$.
3. *Nonnuclear piping:* If a rigorous analysis, including seismic loading condition, is done on a high-energy ASME B31.1 piping, the requirements of the Class 2 and 3 piping mentioned above will apply. If no analysis is performed, breaks are postulated at the following locations:

   a. Terminal ends

   b. At all fittings, welded attachments, and valves

The detailed pipe break design criteria and guidelines are given in the NRC Standard Review Plan No. 3.6.1 and 3.6.2.[51,52]

No-Break Zone. In the design of nuclear power plants, the region of piping in the containment building penetration areas between the isolation valves requires extra protection so that neither the leak-tight integrity of the containment nor the operability of the containment isolation valves is jeopardized. The extra protection consists of the following:

1. Installing special whip restraints, called *isolation restraints,* to mitigate the effects of the postulated pipe breaks located beyond this region
2. Keeping the primary plus secondary stresses and the cumulative usage factor below certain conservative values
3. Holding the piping stress, the isolation valve acceleration, and the stress at the valve-pipe weld below specified limits during a postulated pipe break outside this region
4. Special construction (welding) requirements and in-service inspection procedures

Because of the stringent design requirements, no pipe breaks are assumed to occur in this region. This area of piping is often referred to as the *no-break zone*, the *break exclusion region*, or the *superpipe area*.

No-Break Zone Piping Analysis. An analysis is required to determine the stresses in no-break zone piping and the accelerations of isolation valves due to a postulated break located beyond this region. During the pipe break event, a portion of the piping and the isolation restraints may enter the inelastic region because of the large pipe break loads imposed on the piping system. A static method, or the energy balance method, is acceptable but usually not used because necessary information on the no-break zone such as isolation valve acceleration is impossible to determine. Therefore, a nonlinear dynamic analysis utilizing the direct step-by-step integration method is necessary for the no-break zone analysis.[53] Computer programs based upon the direct integration method with linear elastic and nonlinear inelastic capabilities are often used for this type of analysis.[54,55]

In the analysis, the piping structural model is similar to that described in the subsection "Steam Hammer–Water Hammer Analysis." The nonlinear effects are accounted for by updating the system stiffness matrix at the end of each time step. The integration time step must be short enough to permit a reliable and stable solution. In addition, suitable system damping values should be used to obtain numerical stability. The time-history pipe break forcing function can be calculated by thermal-hydraulic computer programs as described in the subsection "Steam Hammer–Water Hammer Analysis," or obtained from the acceptable simplified method specified in Appendix B of ANSI/ANS 58.2.[56]

Wind Loads. The wind possesses kinetic energy by virtue of the velocity and mass of the moving air. If an obstacle is placed in the path of the wind so that the moving air is stopped or is deflected, then all or part of the kinetic energy of the wind is transformed into the potential energy of pressure.

A piping system which is located outdoors is usually designed to withstand the maximum wind velocity expected during the system operating life.

Dynamic Pressure. The intensity of wind pressure depends on the shape of the obstacle, the angle of incidence of the wind, and the velocity and density of the air.

For standard air (density of the air = 0.07651 lb/ft$_3$, temperature = 59°F), the expression for the wind dynamic pressure could be adapted from Bernoulli's equation for fluid flow as follows[33,57]:

$$p = 0.00256V^2C_D \tag{B4.59}$$

where p = dynamic pressure, lb/ft^2
V = basic wind speed, mi/h
C_D = drag coefficient, dimensionless

For the case of piping under wind loading, Eq. (B4.59) can be rewritten as

$$F = 0.000213V^2C_DD \tag{B4.60}$$

where F = linear dynamic pressure loading on projected pipe length, lb/ft
D = pipe diameter, including insulation, in

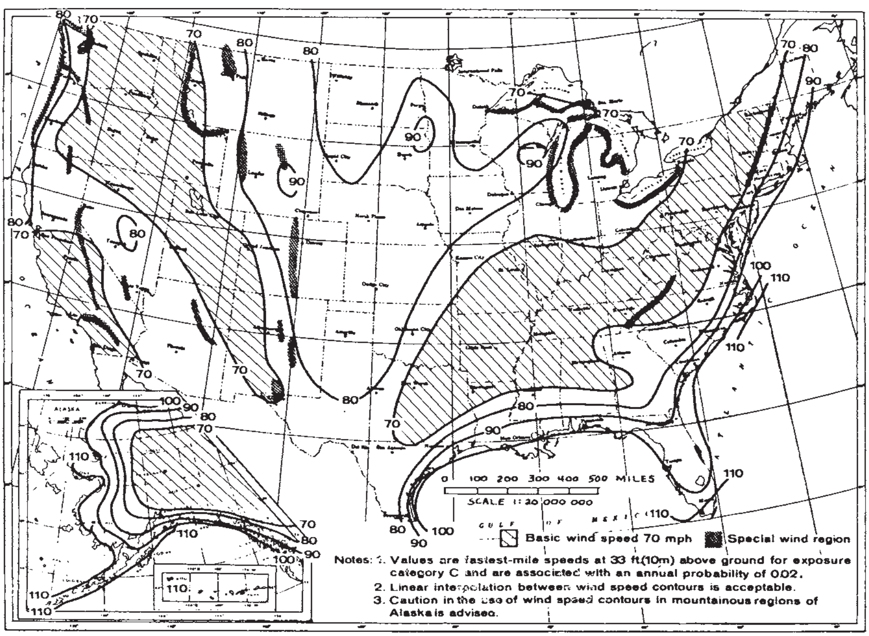


FIGURE B4.14 Basic wind speed (miles per hour). (*ANSI A58.1, 1982. Courtesy of ANSI.*)

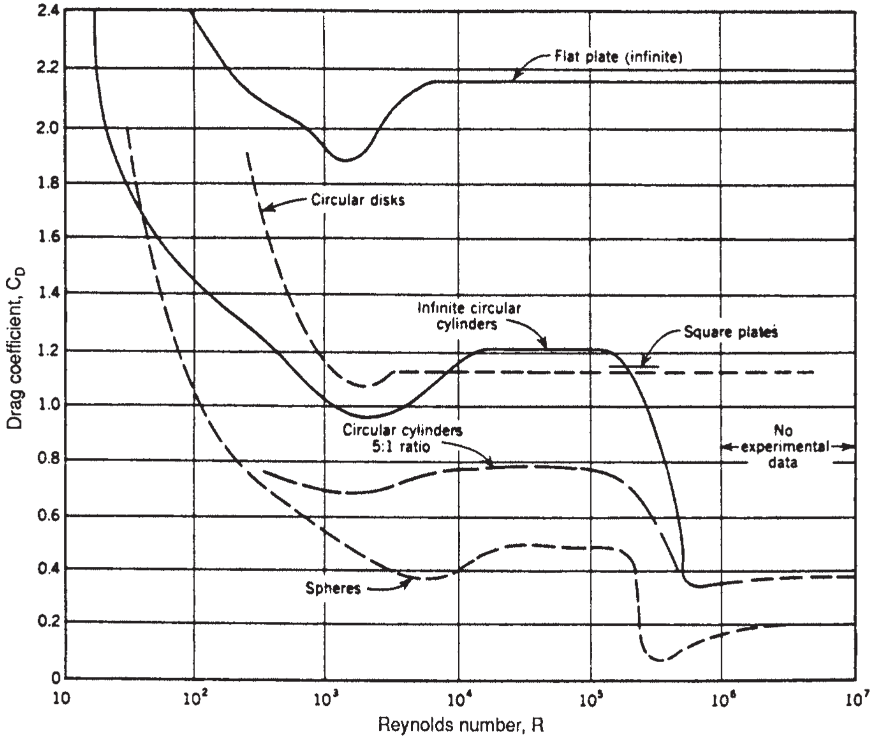


FIGURE B4.15 Drag coefficients for spheres and long cylinders. (*Task Committee on Wind Forces, "Wind Forces."*[30] Courtesy of ASCE.)

Basic Wind Speed. The basic wind speed V is the fastest wind speed at 33 ft above the ground in open terrain with scattered obstructions having heights less than 30 ft, as given in Fig. B4.14 for the United States.[33] The basic wind speed used for design shall be at least 70 mi/h.

Drag Coefficient. The drag coefficient C_D is a function of the shape of the structure and a fluid flow factor called the *Reynolds number.* The Reynolds number R is the ratio of the inertial force to the viscous force which a fluid stream exerts on an object. For standard air, the Reynolds number R could be expressed as

$$R = 780VD \tag{B4.61}$$

The drag coefficient C_D for a cylinder (i.e., a pipe) is given versus the Reynolds number in Fig. B4.15.

Wind Loading Analysis. The piping wind loading analysis is usually performed by a static method. In the analysis, the wind loading F is modeled as a uniform load acting over the projected length of the pipe, parallel to the direction of the wind. Two horizontal directions of wind loads (north-south and east-west) are included in the analysis. The design loads are based on the worst case of the two directions. Similar to the case of earthquake, the wind loading is considered reversing. For load

TABLE B4.8 Gust Response Factor G

Height above ground level, ft	G Exposure A	Exposure B	Exposure C	Exposure D
0–15	2.36	1.65	1.32	1.15
20	2.20	1.59	1.29	1.14
25	2.09	1.54	1.27	1.13
30	2.01	1.51	1.26	1.12
40	1.88	1.46	1.23	1.11
50	1.79	1.42	1.21	1.10
60	1.73	1.39	1.20	1.09
70	1.67	1.36	1.19	1.08
80	1.63	1.34	1.18	1.08
90	1.59	1.32	1.17	1.07
100	1.56	1.31	1.16	1.07
120	1.50	1.28	1.15	1.06
140	1.46	1.26	1.14	1.05
160	1.43	1.24	1.13	1.05
180	1.40	1.23	1.12	1.04
200	1.37	1.21	1.11	1.04
250	1.32	1.19	1.10	1.03
300	1.28	1.16	1.09	1.02
350	1.25	1.15	1.08	1.02
400	1.22	1.13	1.07	1.01
450	1.20	1.12	1.06	1.01
500	1.18	1.11	1.06	1.00

Source: ANSI A58.1, 1982. (*Courtesy of ANSI.*) (1 ft = 0.3048 m)

combination, the wind and the earthquake are assumed to not happen at the same time. A safety factor, the gust response factor G, should also be considered in the analysis. This factor is used to account for the fluctuating nature of wind and its interaction with structures. Its value depends on the exposure categories as shown in Table B4.8, where:

1. *Exposure A:* Large city centers with at least 50 percent of the buildings having a height in excess of 70 ft
2. *Exposure B:* Urban and suburban areas, wooded areas, or other terrain with numerous closely spaced obstructions having the size of single-family dwellings or larger
3. *Exposure C:* Open terrain with scattered obstructions having heights generally less than 30 ft
4. *Exposure D:* Flat, unobstructed coastal areas directly exposed to wind flowing over large bodies of water

METHODS OF ANALYSIS

Cookbook-Type Analysis

The following cookbook-type method is mainly for supporting 2-in and smaller Nuclear Class 2, 3, and B31 piping under gravity, thermal expansion, and seismic loadings. This method is based on standard support span tables. It covers a simplified weight analysis, a simplified thermal analysis, as well as a simplified seismic analysis.

A simplified seismic analysis often requires many pipe supports that are designed to large loads. The cost saving in engineering is offset by increased fabrication and installation cost. The current approach is to analyze the nonseismic piping by simplified methods and all seismic piping by computerized analysis. This greatly reduces the number of required seismic supports and gives an overall cost saving.

Simplified Weight Analysis. A simplified weight analysis is performed by locating the gravity supports based on gravity pipe spans. The maximum gravity pipe spans can be calculated from the following formula:

$$L = \left(\frac{SZ}{1.2W}\right)^{1/2} \tag{B4.62}$$

where L = maximum gravity pipe span, ft
Z = section modulus of pipe, in^3
W = distributed weight, lb/ft
S = pipe stress due to gravity, psi

and the corresponding stress S is

$$S = \frac{M}{Z} = \frac{1.2WL^2}{Z} \tag{B4.63}$$

where M = bending moment, in · lb.

Alternatively, the bending stress in empty pipe may be read from Fig. B4.16, and the bending stress in water-filled pipe from Fig. B4.17. The deflection of empty pipe can be read from Fig. B4.18.

The distributed weight of pipe includes the weight of metal, the weight of pipe contents, and the weight of insulation. Pipe material weights are subject to tolerance of applicable manufacturing specifications.

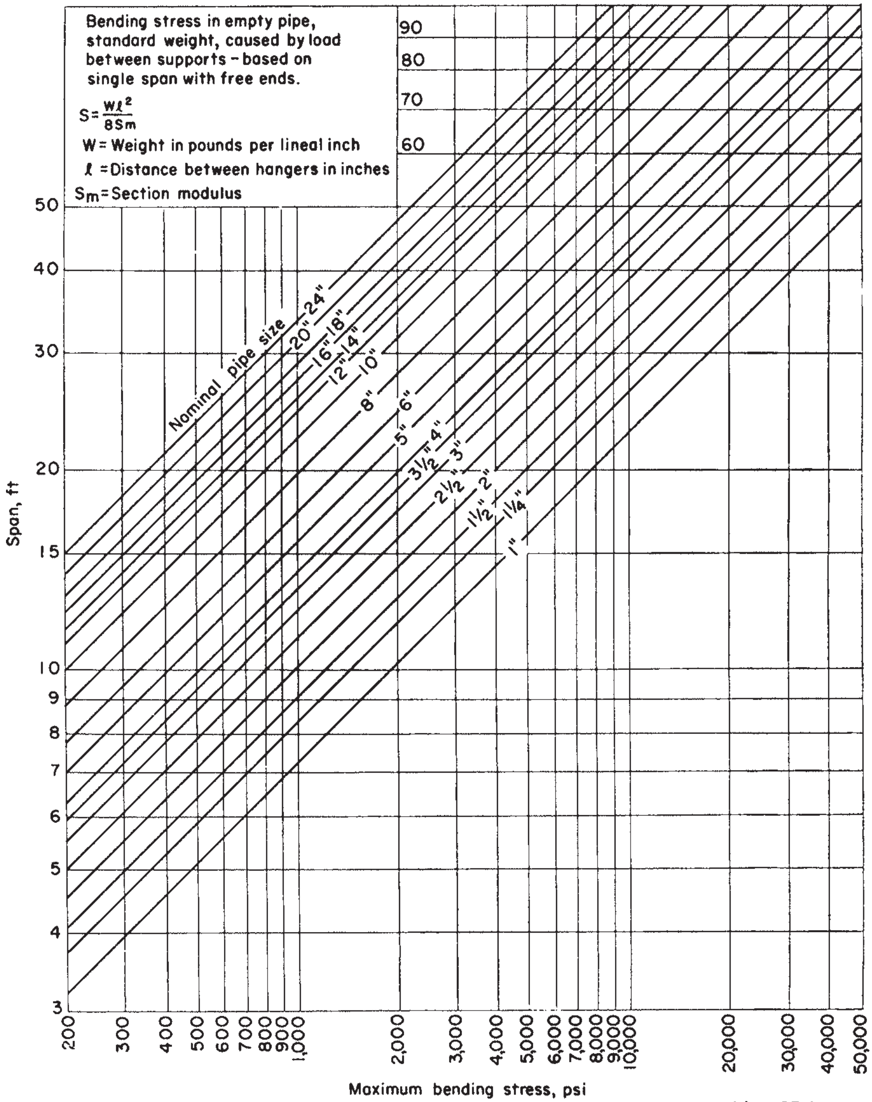
Weights of insulation depend on the composition of insulation material and should be obtained from the insulation manufacturer. Weights of weatherproof protection, if specified, must be added. Insulation thicknesses recommended by insulation manufacturers do not necessarily agree with insulation specifications for a particular job. Insulation specifications should be reviewed prior to development of final weights of piping.

Weights of insulation should be added to weights of flanges, valves, and fittings. Flange, flanged valve, and flanged fitting weights should include weights of bolts and nuts.

Valve weights vary among particular manufacturers' designs and should include weights of electric-motor operators (if any) or other devices which may be specified for particular valves. It is suggested that, wherever possible, valve weights should be obtained from the manufacturer of the particular valves which are to be installed in the piping.

Equation (B4.62) is based on the combination of a simply supported beam model and a fixed-end beam model because the behavior of pipe lies somewhere between these two models.

A table of suggested maximum spans between supports of pipe based on a formula similar to Eq. (B4.62) is given in ASME Codes,[58,59] as shown in Fig. B5.1 of Chap. B5.



1 in = 25.4 mm
1 ft = 0.3048 m
1 psi = 6.8948 kPa

FIGURE B4.16 Bending stress in empty pipe.

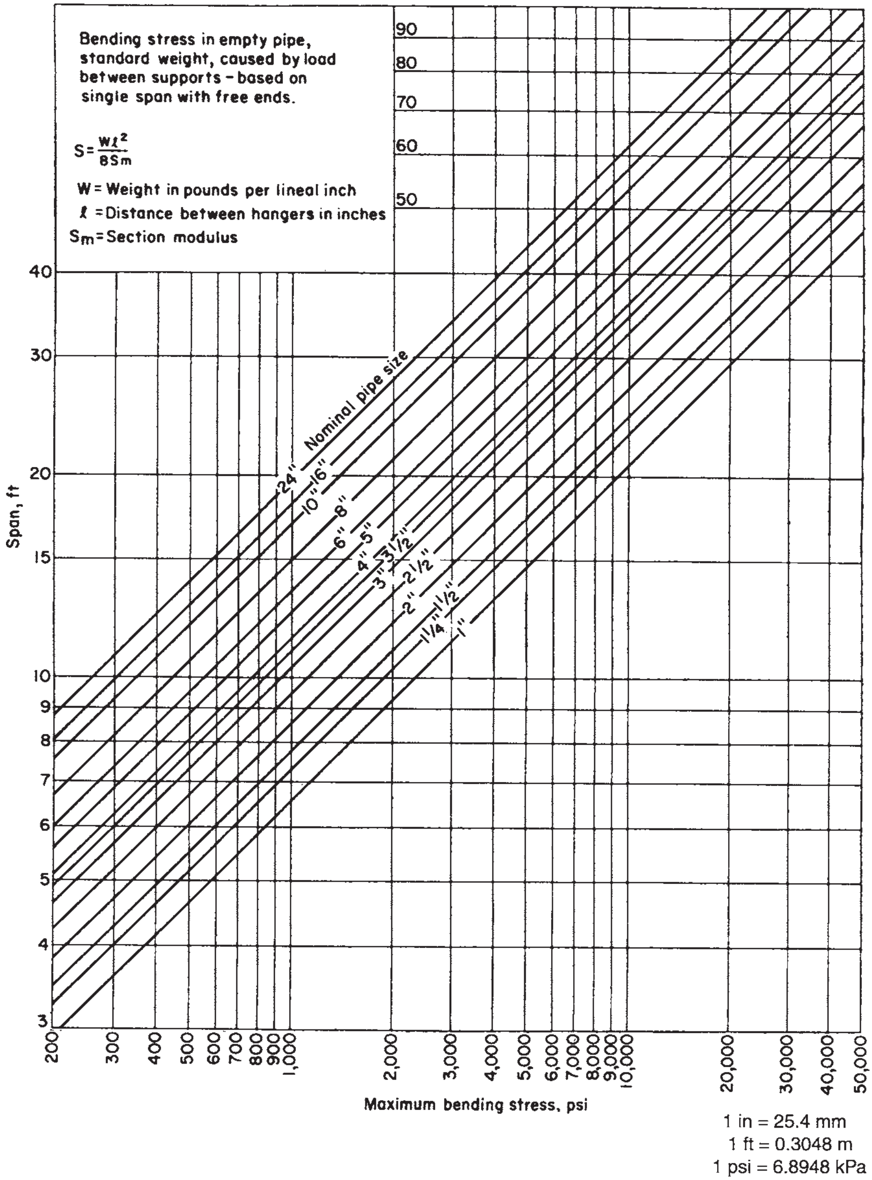


FIGURE B4.17 Bending stress in water-filled pipe.

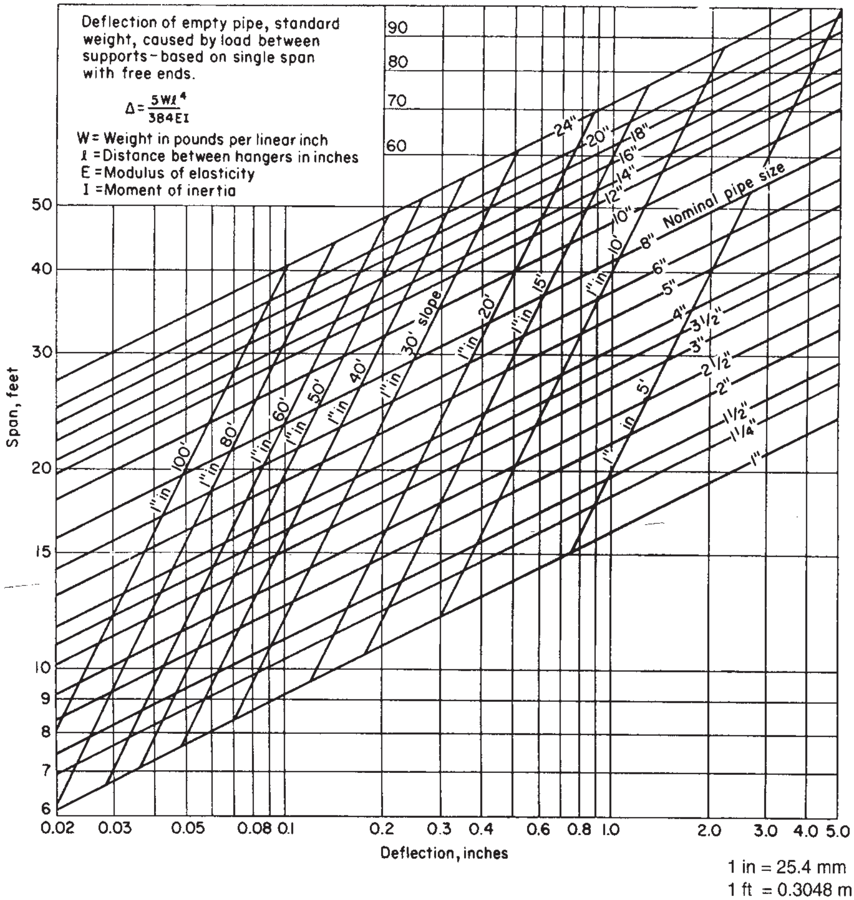


FIGURE B4.18 Deflection of empty pipe.

These spans have been calculated by considering insulated, standard wall thickness and heavier pipe, limited to a maximum stress of 1500 psi (10,350 kPa) and maximum pipe sag of 0.1 in (0.25 mm). For small pipe where socket welds are used, Eq. (B4.63) can be rewritten as

$$S = \frac{0.75iM}{Z} = \frac{1.89WL^2}{Z} \tag{B4.64}$$

and

$$L = \left(\frac{SZ}{1.89W}\right)^{1/2} \tag{B4.65}$$

where i = the stress intensification factor (SIF), 2.1 for socket welds per ASME B31.1 and ASME Section III piping codes.

Figures B5.2 and B5.3 of Chap. B5 give the maximum spans for water and steam, air, or gas filled steel pipe, respectively. These tables are based on a pipe stress S of 2000 psi (13.790 MPa) and a socket weld SIF of 2.1. When these suggested weight spans are adhered to, the stress in the piping system due only to gravity load usually need not be explicitly calculated.

Load Calculation by Weight Balance. The following example is used to illustrate a method by which hanger loadings may be determined. The method consists of locating the center of gravity of the specific piping configuration and then, by equating moments, to determine the resultant loads at particular hangers.

A single-plane bend is shown in Fig. B4.19. Hangers are indicated as *H*-1, *H*-2, *H*-3, and *H*-4. The effects of uniform and concentrated loads are indicated at the points at which these loads act; it is noted that the weight of the 90° bend acts at the centroid of a quarter circle which, in this example, is located 1.8 ft distant from the centerline of the pipe run. The straight pipe length between hangers *H*-3 and *H*-4 is not included in this calculation because it can be analyzed by simple straight-beam theory.

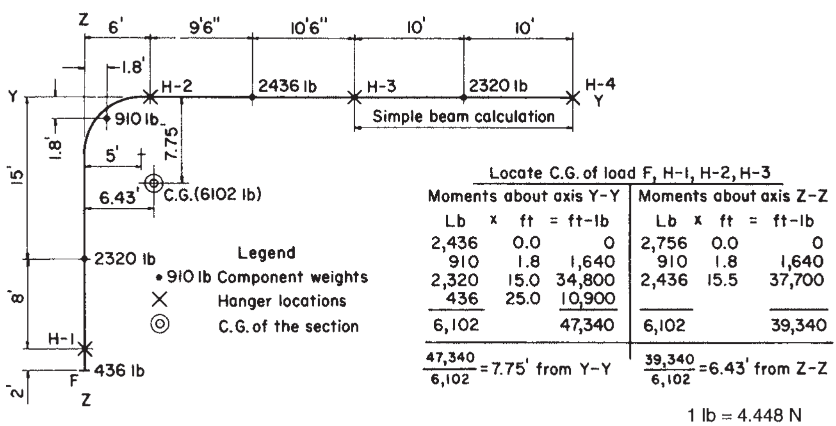
For the piping section which lies between equipment flange *F* and hanger *H*-3, moments are taken about the *Y-Y* and *Z-Z* axes. As an example, let the center of gravity of this configuration be located Y ft from the *Y-Y* axis. Then, from equilibrium considerations, the following equation may be written:

$$2436(0) + 910(1.8) + 2320(15) + 436(25) = 6102Y$$

A solution to this equation results in Y = 7.75 ft.

Similarly, the distance from the *Z-Z* axis to the center of gravity is found to be 6.43 ft.

For convenience, the calculations are made frequently in a tabular fashion as shown in Fig. B4.19.



Locate C.G. of load F, H-1, H-2, H-3

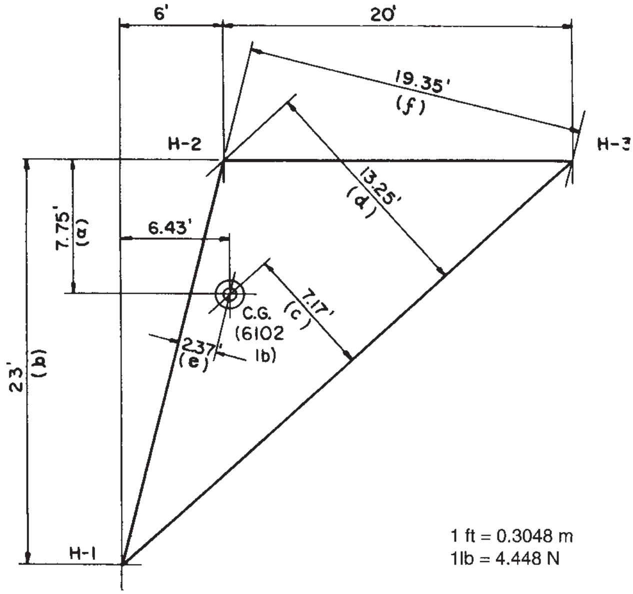
Moments about axis Y-Y: Lb	× ft	= ft-lb	Moments about axis Z-Z: Lb	× ft	= ft-lb
2,436	0.0	0	2,756	0.0	0
910	1.8	1,640	910	1.8	1,640
2,320	15.0	34,800	2,436	15.5	37,700
436	25.0	10,900			
6,102		47,340	6,102		39,340

$\frac{47,340}{6,102}$ = 7.75' from Y-Y $\frac{39,340}{6,102}$ = 6.43' from Z-Z

1 lb = 4.448 N
1 ft = 0.3048 m
1 ft-lb = 1.3561 N-m

FIGURE B4.19 One-line piping diagram for illustration of load calculation by weight balance.

Construct the triangle H-1, H-2, H-3 superimposing location of C.G. and calculate reactions at H-1, H-2 and H-3



$$\text{Reaction at H-1} = \frac{a}{b} \times 6102 = \frac{7.75}{23} \times 6102 = 2055 \text{ lbs}$$

$$\text{H-2} = \frac{c}{d} \times 6102 = \frac{7.17}{13.25} \times 6102 = 3301 \text{ lbs}$$

$$\text{H-3} = \frac{e}{f} \times 6102 \quad \frac{2.37}{19.35} \times 6102 = 746 \text{ lbs}$$

$$F = \quad 0 \text{ lbs}$$

$$6102 \text{ lbs}$$

$$\text{Total load on H-3} = 746 + \frac{2320}{2} = 1906 \text{ lbs}$$

FIGURE B4.20 Hanger load calculations for system of Fig. B4.19. Three hangers with zero reaction at flange *F*.

Let it be now required to determine hanger loadings for the piping configuration of Fig. B4.19 with the stipulation that no load due to weight be imposed on the equipment flange *F*. This is accomplished easily by use of simple geometrical relationships, and the solution is as indicated in Fig. B4.20.

If it were desired to support the piping with two, rather than three, hangers, it would be convenient to eliminate *H*-1 and to relocate *H*-2 to a position at which it would be colinear with the center of gravity and hanger *H*-3. The construction for this arrangement and the associated hanger-load calculations are shown in Fig. B4.21.

In each of the two above cases, one-half of the 2320-lb load between *H*-3 and *H*-4 has been included in the calculations for hanger loading on *H*-3. Thus *H*-4 would be required to support 1160 lb plus, of course, any additional piping load to the right of *H*-4 in Fig. B4.19.

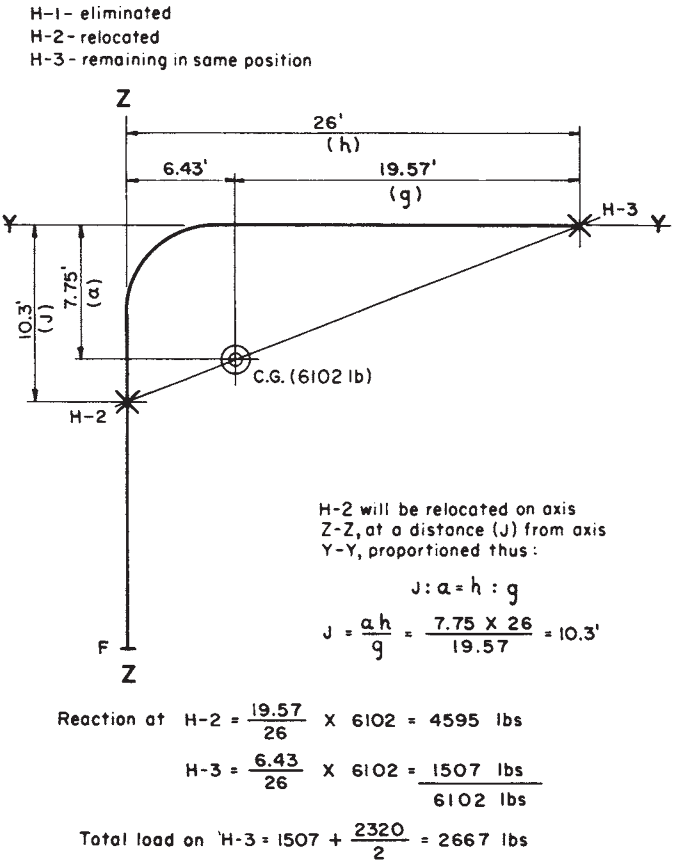


FIGURE B4.21 Hanger load calculations for system of Fig. B4.19 except that one hanger has been eliminated.

Simplified Thermal Expansion Analysis. This simplified analysis is based on the guided cantilever method. The *guided cantilever* is a cantilever beam restrained in such a way that its free end will not rotate when it is deflected in a direction perpendicular to the longitudinal axis of the beam, as shown in Fig. B4.22.

For piping systems under thermal expansion loads, the behavior of the piping approximates that of a guided cantilever. The thermal growth forces the pipe leg to

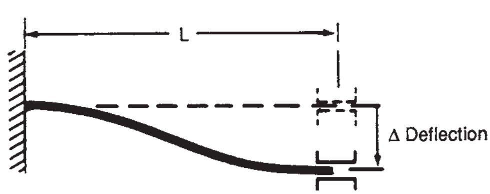


FIGURE B4.22 Guided cantilever.

translate while pipe rotations are restricted by piping continuity. Therefore, this method can be used to check the flexibility of a piping system.

For a guided cantilever, the moment induced by an imposed deflection is

$$M = \frac{6EI\Delta}{L^2} \tag{B4.66}$$

where M = induced moment, in · lb
E = modulus of elasticity, psi
I = moment of inertia, in^4
Δ = deflection, in
L = length of pipe leg perpendicular to deflection, in

The corresponding stress is then

$$S = \frac{iM}{Z} = \frac{6EI\Delta i}{ZL^2} = \frac{3ED\Delta i}{L^2} \tag{B4.67}$$

where S = induced stress, psi
D = outside diameter of pipe, in
Z = section modulus of pipe, in^3
i = stress intensification factor

Solving the equation for the beam length L gives

$$L = \left(\frac{3ED\Delta i}{S}\right)^{1/2} \tag{B4.68}$$

By determining the proper allowable stress and taking into account the appropriate stress intensification factor, Eq. (B4.68) gives an estimate of the minimum allowable offset pipe span L required to sustain a piping thermal movement Δ normal to the piping.

Tables B4.9 and B4.10 give the minimum allowable offset span for steel piping ($E = 27.9 \times 10^6$ psi) with socket welds ($i = 2.1$) and without socket welds, respectively. These tables are based on allowable stresses S of 22,500 psi.

Thermal Movement Calculations. The simplified method shown below is one which gives satisfactory approximations of the piping movements. Whenever differences occur between the approximations and actual movements, the approximation of the movement will always be the greater amount.

Step 1. The piping system of Fig. B4.23 is drawn, and on it are shown all known vertical movements of the piping from its cold to hot, or operating, position. These movements will include those supplied by the equipment manufacturers for the terminal point connections. For the illustrated problem, the following vertical movements are known:

Point A—2 in up, cold to hot
Point B—1 $^1/_6$ in up, cold to hot
Point C—$^1/_8$ in down, cold to hot
H-4—0 in cold to hot

TABLE B4.9 Thermal Expansion Minimum Allowable Offset Span (Feet-Inches), Straight Steel Pipe with Socket Welds

Therm. expan., in	Pipe size, NPS (DN)				
	½ (15)	¾ (20)	1 (25)	1½ (40)	2 (50)
0.10	2-1¾	2-4¾	2-8	3-2½	3-7
0.15	2-7½	2-11	3-3¼	3-11¼	4-4¾
0.20	3-0¼	3-4½	3-9¼	4-6½	5-1
0.25	3-4½	3-9¼	4-2¾	5-1	5-8
0.30	3-8½	4-1½	4-7½	5-6¾	6-2½
0.35	4-0	4-5½	5-0	6-0	6-8½
0.40	4-3¼	4-9¼	5-4	6-5	7-2¼
0.45	4-6¼	5-0¾	5-8	6-9¾	7-7¼
0.50	4-9¼	5-4	5-11¾	7-2¼	8-0¼
0.60	5-2¾	5-10¼	6-6½	7-10¼	8-9½
0.70	5-7¾	6-3¾	7-0¾	8-6	9-6
0.80	6-0½	6-9	7-6¾	9-1	10-1¾
0.90	6-4¾	7-2	8-0¼	9-7½	10-9¼
1.00	6-9	7-6½	8-5¼	10-1¾	11-4¼
1.10	7-1	7-11	8-10¼	10-7¾	11-10¾
1.20	7-4¾	8-3¼	9-3	11-1½	12-5¼
1.50	8-3¼	9-3	10-4¼	12-5¼	13-10¾
1.75	8-11¼	9-11¾	11-2	13-5¼	15-0¼
1.90	9-3¾	10-4¾	11-7¾	14-0	15-7¾
2.00	9-6½	10-8	11-11¼	14-4¼	16-0¾
2.50	10-8	11-11¼	13-4¼	16-0¾	17-11¼
3.00	11-8¼	13-0¾	14-7¼	17-7	19-8

1 in = 25.4 mm
1 ft = 0.3048 m

The operating temperature of the system is given as 1050°F (566°C), and the coefficient of expansion for low-chrome steel at 1050°F (566°C) is 0.0946 in/ft.

The movements at points *D* and *E* are calculated by multiplying the coefficient of expansion by the vertical distance of each point from the position of zero movement on the riser *DE:*

$$55\text{ ft} \times 0.0946\text{ in/ft} = 5.20\text{ in } \textit{up} \text{ at } D$$

$$20\text{ ft} \times 0.0946\text{ in/ft} = 1.89\text{ in } \textit{down} \text{ at } E$$

Step 2. A simple drawing is made of the piping between two adjacent points of known movement, extending the piping into a single plane as shown for the portion of the system between *A* and *D*.

The vertical movement at any hanger location will be proportional to its distance from the endpoints:

$$\Delta_1 = {}^4\!/_{31} \times 3.20$$

$$\Delta_1 = 0.41\text{ in}$$

TABLE B4.10 Thermal Expansion Minimum Allowable Offset Span (Feet-Inches), Straight Steel Pipe, No Socket Weld

Therm. expan., in	Pipe size, NPS (DN) ½ (15)	¾ (20)	1 (25)	1½ (40)	2 (50)
0.10	1-5¾	1-8	1-10¼	2-2¾	2-5¾
0.15	1-9¾	2-0¼	2-3¼	2-8¾	3-0½
0.20	2-1¼	2-4	2-7¼	3-1¾	3-6¼
0.25	2-4	2-7¼	2-11¼	3-6¼	3-11
0.30	2-6¾	2-10¼	2-2½	3-10¼	4-3½
0.35	2-9¼	3-1	3-5½	4-1¾	4-7¾
0.40	2-11½	3-3¾	3-8¼	4-5¼	4-11¾
0.45	3-1½	3-6	3-11	4-8½	5-3
0.50	3-3¾	3-8¼	4-1¾	4-11¾	5-6½
0.60	3-7½	4-0½	4-6¼	5-5¼	6-1
0.70	3-10¾	4-4½	4-10½	5-10½	6-6¾
0.80	4-2¼	4-8	5-2¾	6-3¼	7-0¼
0.90	4-5	4-11½	5-6½	6-7¾	7-5¼
1.00	4-8	5-1½	5-10	7-0¾	7-10¼
1.10	4-10¾	5-5¾	6-1¾	7-4¼	8-2¾
1.20	5-1¼	5-8½	6-4¾	7-8¼	8-7
1.50	5-8½	6-4¾	7-1¾	8-7	9-7¼
1.75	6-2¼	6-10¾	7-8½	9-3½	10-4½
1.90	6-5¼	7-2¼	8-0½	9-8	10-9¾
2.00	6-7¼	7-4½	8-3	9-11	11-1
2.50	7-4¾	8-3¼	9-3	11-1½	12-5
3.00	8-1¼	9-0¾	10-1¾	12-2¼	13-7½

1 in = 25.4 mm
1 ft = 0.3048 m

The vertical movement at H-1 = 0.41 in + 2 in:

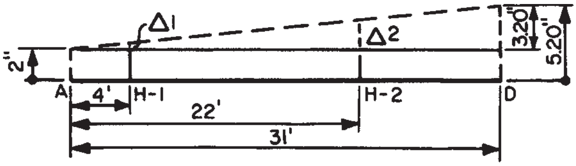
$$\Delta H\text{-}1 = 2.41 \text{ in up}$$

$$\Delta_2 = {}^{22}\!/_{31} \times 3.20$$

$$\Delta_2 = 2.27 \text{ in}$$

The vertical movement at H-2 = 2.27 in + 2 in:

$$\Delta H\text{-}2 = 4.27 \text{ in up}$$

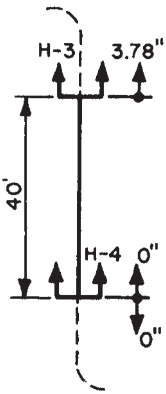


(*a*)

Step 3. To calculate the vertical movement at H-3, multiply its distance from *H*-4 by the coefficient of expansion:

$$40 \text{ ft} \times 0.0946 \text{ in/ft} = 3.78 \text{ in}$$

$$\Delta H\text{-}3 = 3.78 \text{ in up}$$



(*b*)

Step 4. The next section of pipe on which there are two points of known movement is the length *E-J*. The movement at *E* was calculated as 1.89 in down:

The movement at J is equal to the movement at the terminal point *C* (1/8 in down) plus the amount of expansion of the leg *C-J:*

$$\Delta J = 0.125 \text{ in} + 3.5 \text{ ft} \times 0.0946 \text{ in/ft}$$

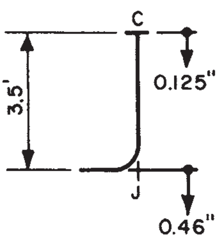
$$\Delta J = 0.46 \text{ in down}$$

$$\Delta_7 = {}^{3.5}/_{42} \times 1.43 = 0.12 \text{ in}$$

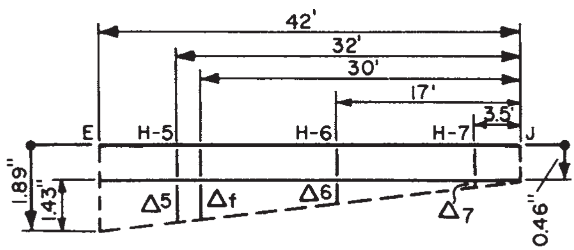
$$\Delta H\text{-}7 = 0.12 \text{ in} + 0.46 \text{ in}$$

$$\Delta H\text{-}7 = 0.58 \text{ in down}$$

$$\Delta_6 = {}^{17}/_{42} \times 1.43 = 0.58 \text{ in}$$



(*a*)



(*b*)

$$\Delta H\text{-}6 = 0.58 + 0.46 \text{ in}$$

$$\Delta H\text{-}6 = 1.04 \text{ in down}$$

$$\Delta f = {}^{30}/_{42} \times 1.43 = 1.02 \text{ in}$$

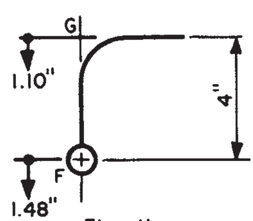
$$\Delta F = 1.02 + 0.46$$

$$\Delta F = 1.48 \text{ in down}$$

$$\Delta_5 = {}^{32}/_{42} \times 1.43 = 1.09 \text{ in}$$

$$\Delta H\text{-}5 = 1.09 + 0.46$$

$$\Delta H\text{-}5 = 1.55 \text{ in down}$$



Elevation

(*c*)

Step 5. In the section *G-H*, the movement at *G* is equal to the movement at *F minus* the expansion

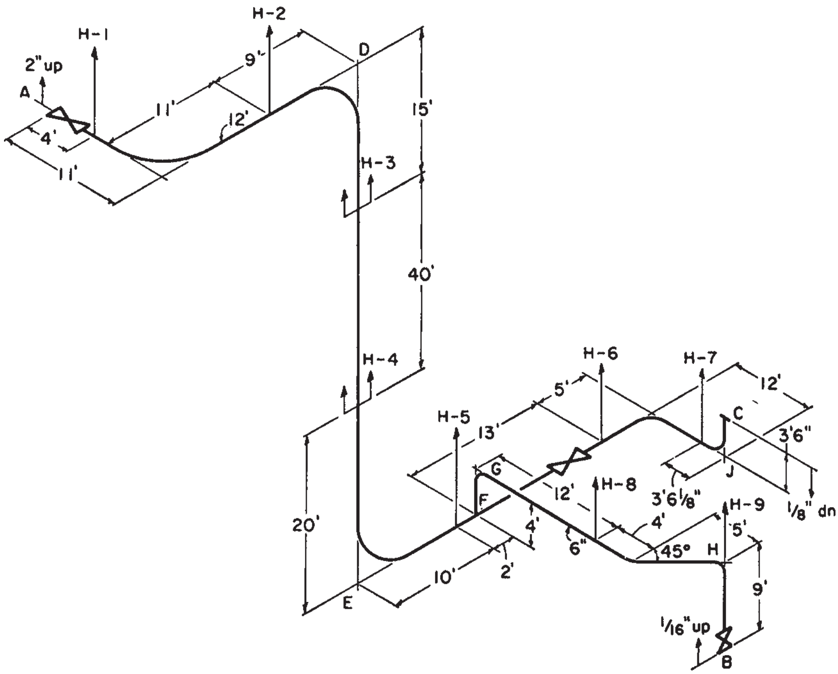


FIGURE B4.23 One-line piping diagram for calculation of hanger movements. Points *A*, *B*, and *C* are equipment connections. *H*-1, *H*-2, and so on, represent hanger locations.

of the leg *GF:*

$$\Delta G = 1.48 \text{ in down} - 4 \text{ ft} \times 0.0946 \text{ in/ft}$$

$$\Delta G = 1.10 \text{ in down}$$

The movement at *H* is equal to the movement of the terminal point *B* (1/16 in up) *plus* the expansion of the leg *B-H:*

$$\Delta H = 0.0625 \text{ in up} + 9 \text{ ft} \times 0.0946 \text{ in/ft}$$

$$\Delta H = 0.91 \text{ in up}$$

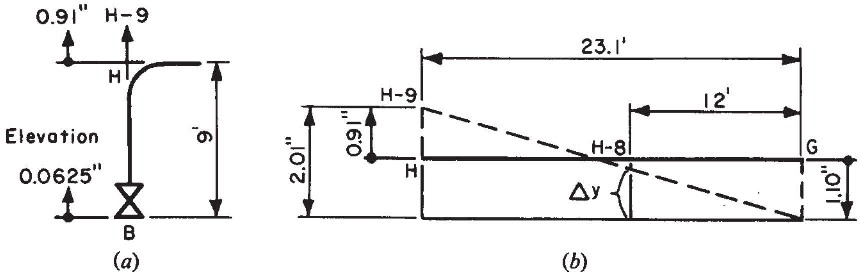
Since *H-9* is located at point *H*,

$$\Delta H\text{-}9 = \Delta H = 0.91 \text{ in up}$$

$$\Delta_y = \frac{12 \times 2.01}{23.1} = 1.04 \text{ in}$$

$$\Delta H\text{-}8 = 1.10 - 1.04$$

$$\Delta H\text{-}8 = 0.60 \text{ in down}$$



After calculating the movement at each hanger location it is often helpful, for easy reference when selecting the appropriate type hanger, to make a simple table of hanger movements.

Hanger number	Movement, in
H-1	2.41 up
H-2	4.27 up
H-3	3.78 up
H-4	0
H-5	1.55 down
H-6	1.04 down
H-7	0.58 down
H-8	0.06 down
H-9	0.91 up

1 in = 25.4 mm

Calculation of Hanger Loads. A 6-in medium-temperature steam piping system is shown in Fig. B4.24. Terminal movements at equipment flanges *A* and *B* are indicated; dimensions of system components and physical data are also given. It is required to determine hanger loadings and also to determine movements at each of the hangers *H*-1 through *H*-7.

It is noted that hanger *H*-3 on the vertical leg has been located 20 ft (6.0 m) above the lower horizontal pipe run. Calculations would indicate that the center of gravity of the vertical leg is 16.16 ft (5.0 m) above the lower horizontal run. It would not be desirable to place the hanger at the center of gravity because the hanger would then act as a pivot point and would not resist sway. If the hanger *H*-3 were placed below the center of gravity, an unstable turnover condition would result. The most desirable location is above the center of gravity; hanger *H*-3 has thus been placed arbitrarily a distance of 20 ft (6.0 m) above the lower horizontal piping run.

Starting with equipment flange *A*, the system is broken up into component parts between hangers and hanger reactions are calculated. The procedure is indicated in Figs. B4.25*a* to B4.25*g*, and the results are listed in Table B4.11. Hanger deflections, or movements, are determined as shown in Figs. B4.26*a* and B4.26*b*.

Simplified Seismic Analysis. A simplified seismic analysis utilizing simple beam formulas and response spectrum curves is given here. The maximum support spacings are selected

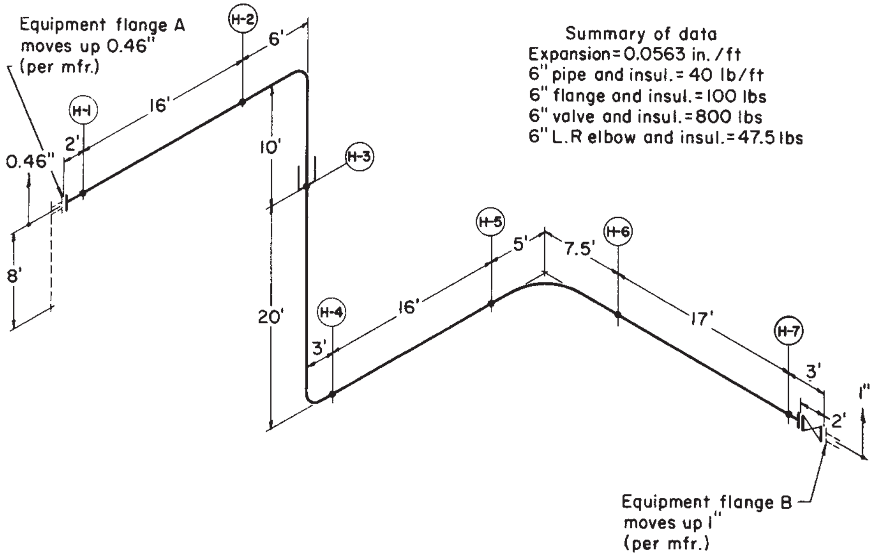


FIGURE B4.24 One-line piping diagram for calculation of hanger loadings and deflections.

from Tables B4.12 to B4.14 so that the fundamental frequency of the span is in the rigid range of the response spectrum.

These seismic spans are based on the following formula:

$$L = \left(\frac{1}{12}\right)\left(\frac{\pi}{2f}\right)^{1/2}\left(\frac{12gEI}{W}\right)^{1/4} \tag{B4.69}$$

where L = maximum seismic spacing, ft
f = desired frequency, cycles/s
g = 386 in/s^2
E = modulus of elasticity, psi
I = moment of inertia, in^4
W = distributed weight, lb/ft

For a system with seismic supports designed in the rigid range, the seismic acceleration of the system is low and consequently the design loads for the system decrease.

The corresponding seismic stress is then

$$S = 0.75i(12)\left(\frac{WL^2}{8Z}\right)(1.5G) \tag{B4.70}$$

where Z = section modulus of pipe, in^3
G = seismic acceleration (OBE or SSE) in gs
i = stress intensification factor

The number 1.5 in Eq. (B4.70) is a factor to account for the contribution from the higher modes.[47,60]

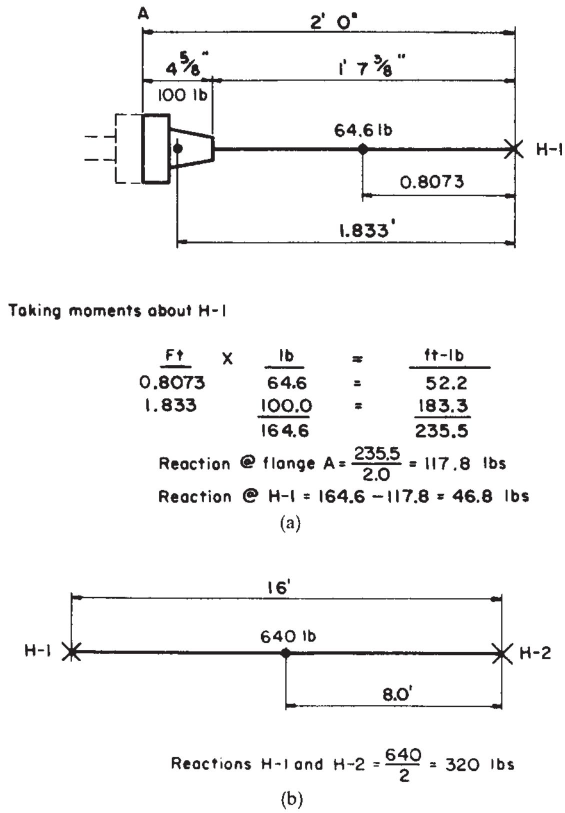


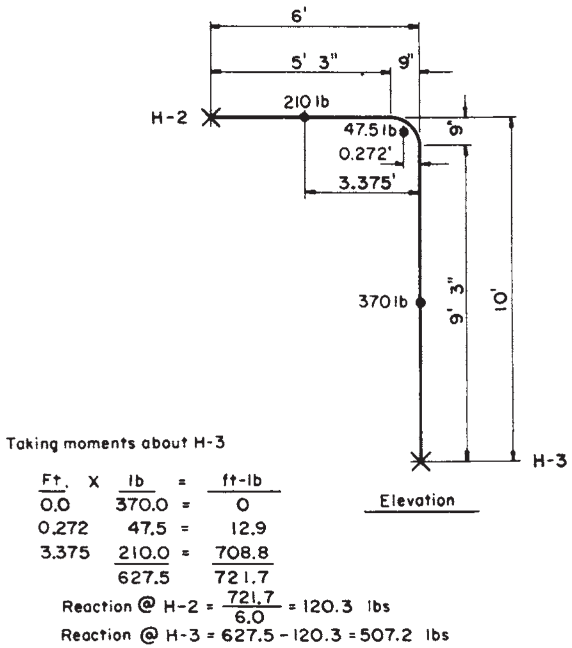
FIGURE B4.25 *(a)* Distribution of weight between equipment flange *A* and *H*-1. *(b)* Distribution of weight between *H*-1 and *H*-2.

Computerized Method

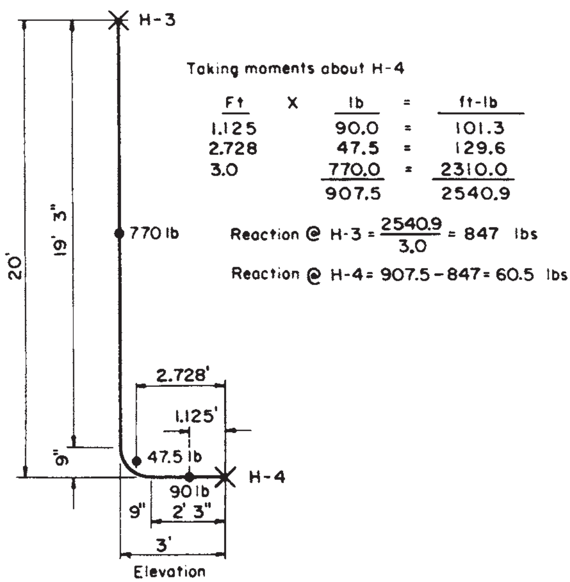
Types of Computer Programs. The microcomputer has become the daily tool and workstation of the piping stress analyst. Files which contain data for piping stress analysis are created, edited, and saved at this workstation. These files are later transferred to the mini- or mainframe computer for the calculation of piping stresses and support loads.

Most of the computer programs for piping stress analysis such as ADLPIPE, NUPIPE, and SUPERPIPE were developed for use on mainframe computers. With the introduction of many powerful microcomputers in the mid-1980s, microcomputer-based programs for piping stress analysis were also developed such as AU-TOPIPE and CAESAR II. Some of these new programs are menu driven and user friendly. Refer to Appendix E9. They help save engineering time and cost. In general, these computer programs may be divided into four classes:

1. Programs that can perform pressure, thermal expansion, deadweight, and external forces (e.g., wind) analyses for ASME Section III, Class 2, 3, ASME B31.1, B31.3,



(c)



(d)

FIGURE B4.25 *(c)* Distribution of weight between *H*-2 and *H*-3. *(d)* Distribution of weight between *H*-3 and *H*-4.

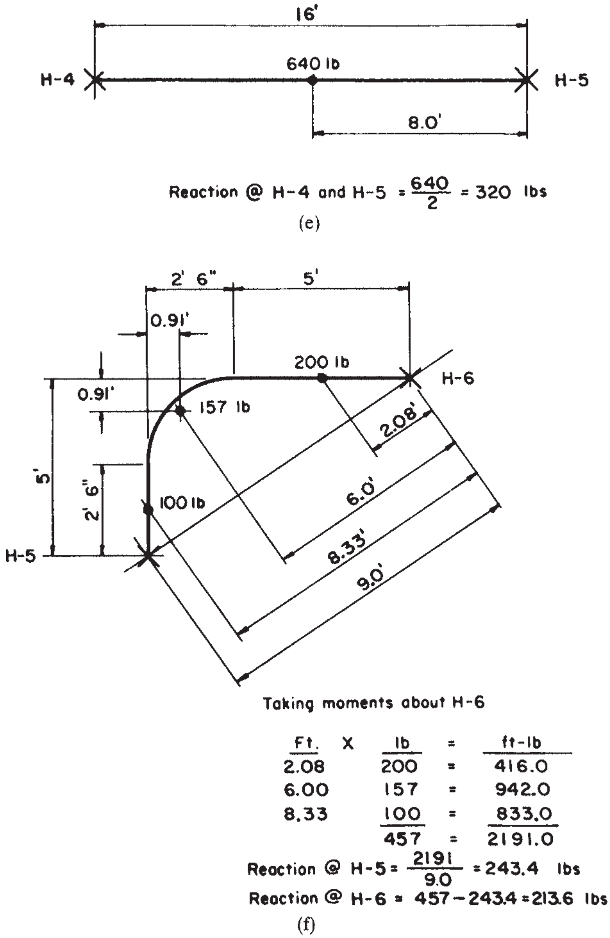
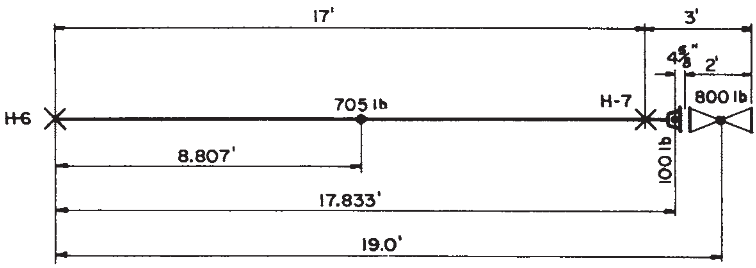


FIGURE B4.25 *(e)* Distribution of weight between *H*-4 and *H*-5. (f) Distribution of weight between *H*-5 and *H*-6.

B31.4, B31.5, B31.8, NEMA, API-610, and API-617 piping. Programs such as TRIFLEX, AUTOPIPE, and CAESAR II are in this class. (AUTOPIPE and CAESAR II have response spectrum and SAM analysis capability. However, there is a limit on the number of analyses which can be performed in the same computer run because of the memory capability of microcomputers.)

2. Programs that can perform seismic, independent support motion, thermal transient, and time-history analyses in addition to those mentioned in item 1 for ASME Section III, Class 1, 2, 3, ASME B31.1, and B31.3 piping. Programs such as ADLPIPE, ME101, NUPIPE, PIPESD, and SUPERPIPE are in this class.
3. General-purpose programs, such as ANSYS. ANSYS is a general-purpose finite element analysis program which can perform static and dynamic analysis; elastic



Taking moments about H-6

Ft	X	lb	=	ft-lb
8.807		705	=	6210
17.833		100	=	1783
19.0		800	=	15200
		1605		23193

$$\text{Reaction @ H-7} = \frac{23193}{17.0} = 1364 \text{ lbs}$$

$$\text{Reaction @ H-6} = 1605 - 1364 = 241 \text{ lbs}$$

FIGURE B4.25 (*g*) Distribution of weight between *H*-6 and *H*-7 to maintain zero reaction on flange *B*.

TABLE B4.11 Summary of Hanger Loadings

Hanger mark	Reactions, lb							Hanger load, lb
	A to *H*-1	*H*-1 to *H*-2	*H*-2 to *H*-3	*H*-3 to *H*-4	*H*-4 to *H*-5	*H*-5 to *H*-6	*H*-6 to *H*-7	
Flange *A*	117.8	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	117.8
H-1	46.8	320.0	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	366.8
H-2	· · · ·	320.0	120.3	· · · ·	· · · ·	· · · ·	· · · ·	440.3
H-3	· · · ·	· · · ·	507.2	847.0	· · · ·	· · · ·	· · · ·	1,354.2
H-4	· · · ·	· · · ·	· · · ·	60.5	320.0	· · · ·	· · · ·	380.5
H-5	· · · ·	· · · ·	· · · ·	· · · ·	320.0	243.4	· · · ·	563.4
H-6	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	213.6	241.0	454.6
H-7	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	1364.0	1,364.0
Flange *B*	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	· · · ·	0.0	0.0

1 lb = 4.448 N

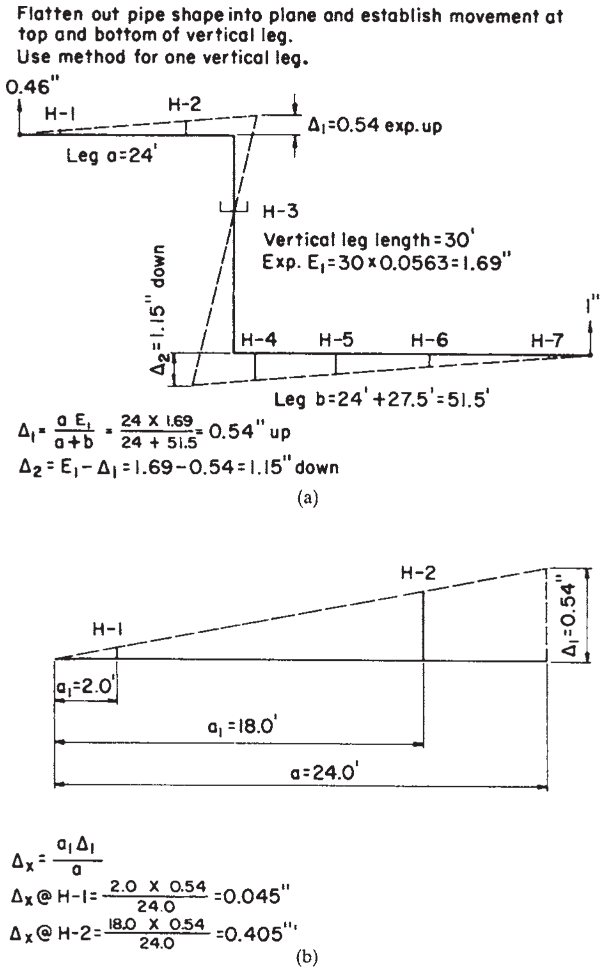


FIGURE B4.26 (a) Deflections of vertical leg of Fig. B4.24. (b) Determination of deflections at *H*-1 and *H*-2 of Fig. B4.24.

and plastic analysis; steady-state and transient heat transfer; steady-state fluid flow analyses; and nonlinear time-history analyses. There are 40 different finite elements available for static and dynamic analysis. Dynamic analyses can be performed either by modal superposition or direct integration.[54]

4. Specialized programs such as PIPERUP. PIPERUP performs nonlinear elasticplastic analyses of piping systems subjected to concentrated static or dynamic time-history forcing functions. These forces result from fluid jet thrusts at the location of a postulated break in high-energy piping. PIPERUP is an adaptation of the finite element method to the specific requirements of pipe rupture analysis.[55]

TABLE B4.12 Maximum Support Spacing for Seismic Stress for a Frequency of 20 Hz (Feet -Inches), Steel Pipe

			Antisweat*		Calcium silicate*					Reflective*
NPS (DN)	Pipe sch.	No insul.	½ in	1 in	½ in	1 in	1½ in	2 in	2½ in	4 in
½ (15)	40	5-4	4-11	4-8	4-9	4-5	4-2	3-10	3-8	3-3
	80	5-4	4-11	4-9	4-10	4-6	4-3	4-0	3-9	3-5
	160	5-3	4-11	4-9	4-10	4-6	4-3	4-0	3-10	3-5
¾ (20)	40	6-0	5-7	5-4	5-5	5-1	4-10	4-7	4-4	3-11
	80	6-0	5-7	5-5	5-6	5-3	4-11	4-8	4-5	4-1
	160	5-11	5-7	5-5	5-6	5-3	5-0	4-9	4-7	4-2
1 (25)	40	6-9	6-4	6-2	6-3	5-11	5-8	5-4	5-1	4-8
	80	6-9	6-5	6-3	6-4	6-0	5-9	5-6	5-3	4-10
	160	6-8	6-5	6-3	6-4	6-1	5-10	5-7	5-5	5-0
1½ (40)	40	8-0	7-8	7-6	7-7	7-4	7-1	6-9	6-7	6-1
	80	8-1	7-10	7-8	7-9	7-6	7-3	7-0	6-9	6-4
	160	8-1	7-10	7-9	7-9	7-7	7-4	7-2	6-11	6-7
2 (50)	40	8-11	8-7	8-5	8-6	8-3	8-0	7-9	7-6	7-1
	80	9-1	8-9	8-8	8-8	8-6	8-3	8-0	7-10	7-5
	160	9-1	8-10	8-9	8-9	8-7	8-5	8-3	8-1	7-8

* Insulation type. 1 in = 25.4 mm, 1 ft = 0.3048 m

TABLE B4.13 Maximum Support Spacing for Seismic Stress for a Frequency of 20 Hz (Feet -Inches), Steel Pipe

			Antisweat*		Calcium silicate*					Reflective*
NPS (DN)	Pipe sch.	No insul.	½ in	1 in	½ in	1 in	1½ in	2 in	2½ in	4 in
½ (15)	40	4-10	4-4	4-2	4-3	4-0	3-8	3-6	3-3	2-11
	80	4-9	4-5	4-3	4-4	4-0	3-9	3-7	3-4	3-0
	160	4-8	4-5	4-3	4-4	4-1	3-10	3-7	3-5	3-1
¾ (20)	40	5-4	5-0	4-9	4-11	4-7	4-4	4-1	3-10	3-6
	80	5-4	5-0	4-10	4-11	4-8	4-5	4-2	4-0	3-7
	160	5-3	5-0	4-10	4-11	4-8	4-6	4-3	4-1	3-9
1 (25)	40	6-0	5-8	5-6	5-7	5-4	5-0	4-10	4-7	4-2
	80	6-0	5-9	5-7	5-8	5-5	5-2	4-11	4-9	4-4
	160	5-11	5-8	5-7	5-8	5-5	5-3	5-0	4-10	4-6
1½ (40)	40	7-2	6-10	6-9	6-9	6-6	6-4	6-1	5-10	5-5
	80	7-3	7-0	6-10	6-11	6-8	6-6	6-3	6-1	5-8
	160	7-3	7-0	6-11	6-11	6-9	6-7	6-5	6-2	5-10
2 (50)	40	8-0	7-8	7-7	7-7	7-4	7-2	6-11	6-9	6-4
	80	8-1	7-10	7-9	7-9	7-7	7-5	7-2	7-0	6-7
	160	8-1	7-11	7-10	7-10	7-8	7-6	7-4	7-2	6-10

* Insulation type. 1 in = 25.4 mm, 1 ft = 0.3048 m

TABLE B4.14 Maximum Support Spacing for Seismic Stress for a Frequency of 20 Hz (Feet -Inches), Steel Pipe

			Antisweat*		Calcium silicate*					Reflective*
NPS (DN)	Pipe sch.	No insul.	½ in	1 in	½ in	1 in	1½ in	2 in	2½ in	4 in
½ (15)	40	4-2	3-9	3-7	3-8	3-5	3-2	3-0	2-9	2-6
	80	4-1	3-10	3-8	3-9	3-5	3-3	3-1	2-10	2-7
	160	4-1	3-10	3-8	3-9	3-6	3-4	3-1	2-11	2-8
¾ (20)	40	4-7	4-4	4-1	4-3	3-11	3-9	3-6	3-4	3-0
	80	4-7	4-4	4-2	4-3	4-0	3-10	3-7	3-5	3-1
	160	4-6	4-4	4-2	4-3	4-0	3-11	3-8	3-6	3-3
1 (25)	40	5-2	4-11	4-9	4-10	4-7	4-4	4-2	3-11	3-7
	80	5-2	5-0	4-10	4-11	4-8	4-5	4-3	4-1	3-9
	160	5-1	4-11	4-10	4-11	4-8	4-6	4-4	4-2	3-11
1½ (40)	40	6-2	5-11	5-10	5-10	5-7	5-6	5-3	5-0	4-8
	80	6-3	6-1	5-11	6-0	5-9	5-7	5-5	5-3	4-11
	160	6-3	6-1	6-0	6-0	5-10	5-8	5-7	5-4	5-0
2 (50)	40	6-11	6-8	6-7	6-7	6-4	6-2	6-0	5-10	5-6
	80	7-0	6-9	6-8	6-8	6-7	6-5	6-2	6-1	5-8
	160	7-0	6-10	6-9	6-9	6-8	6-6	6-4	6-2	5-11

* Insulation type. 1 inch = 25.4 mm, 1 ft = 0.0348 m

Method of Analysis. The piping system is modeled as a series of masses connected by massless springs having the properties of the piping. The mathematical model should include the effects of piping geometry changes, elbow flexibilities, concentrated weights, changes in piping cross sections, and any other parameters affecting the stiffness matrix of the model. Mass point spacing should follow the guidelines specified above. Valves should be modeled as lumped masses at valve body and operator, with appropriate section properties for valve body and valve topworks. Rigid supports, snubbers, springs, and equipment nozzles should be modeled with appropriate spring rates in particular degree of freedom. Stress intensification factors should be input at the appropriate locations (elbows, tees, branch connections, welds, etc.). Piping distributed weight should include pipe weight, insulation weight, and entrained fluid weight.

Once an accurate model is developed, the loading conditions are applied mathematically:

1. Statically applied loads (deadweight, wind loads, pressure thrust, etc.)
2. Thermal expansion
3. Statically applied boundary condition displacements (seismic anchor movement, LOCA containment displacement, etc.)
4. Response spectrum analysis (seismic, etc.)
5. Dynamically applied boundary condition displacements (LOCA motion, etc.)
6. Dynamically applied forcing functions (steam hammer, etc.)

The results of the analyses should be examined in order to determine if all allowables are met (i.e., piping stress, valve acceleration, nozzle loads, etc.). The loads must be combined using the appropriate load combinations and submitted to structural designers for their analysis.

PROCEDURES FOR THE DESIGN OF RESTRAINED UNDERGROUND PIPING

This section is reproduced by the courtesy of ASME B31.1.

Foreword

The B31.1 Code contains rules governing the design, fabrication, materials, erection, and examination of power piping systems. Experience over the years has demonstrated that these rules may be conservatively applied to the design and analysis of buried piping systems. However, the ASME B31.1 rules were written for piping suspended in open space, with the supports located at local points on the pipe. Buried piping, on the other hand, is supported, confined, and restrained continuously by the passive effects of the backfill and the trench bedding. The effects of continuous restraint cannot be easily evaluated by the usual methods applied to exposed piping, since these methods cannot easily accommodate the effects of bearing and friction at the pipe/soil interface. Accordingly, this section has been prepared to illustrate and clarify the application of B31.1 Code rules to restrained buried piping.

All components in the buried piping system must be given consideration, including the building penetrations, branches, bends, elbows, flanges, valves, grade penetrations, and tank attachments. It is assumed that welds are made in accordance with the B31.1. Code and that appropriate corrosion protection procedures are followed for buried piping.

This section provides analytic and nomenclature definition figures to assist the designer, and is not intended to provide actual design layout.

Scope

The scope of this section is confined to the design of buried piping as defined in Pa+ expansion in buried piping affects the forces, the resulting bending moments and stresses throughout the buried portions of the system, particularly at the anchors, building penetrations, buried elbows and bends, and branch connections, and it is the designer's responsibility to consider these forces. This section, however, deals only with the buried portions of the system, and not the complete system.

The design and analysis of buried piping requires that careful attention be paid to:

1. All loads acting on the system
2. The forces and the bending moments in the piping and piping components resulting from the loads
3. The loading and stress criteria
4. General design practices

Definitions

Confining Pressure the pressure imposed by the compacted backfill and overburden on a buried pipe. Confining pressure is assumed to act normal to the pipe circumference.

Flexible Coupling a piping component that permits a small amount of axial or angular movement while maintaining the pressure boundary

Friction the passive resistance of soil to axial movement. Friction at the pipe/soil interface is a function of confining pressure and the coefficient of friction between the pipe and the backfill material. Friction forces exist only where there is actual or impending slippage between the pipe and soil.

Influence Length that portion of a transverse pipe run which is deflected or "influenced" by pipe thermal expansion along the axis of the longitudinal run

Modulus of Subgrade Reaction the rate of change of soil bearing stress with respect to compressive deformation of the soil. It is used to calculate the passive spring rate of the soil.

Penetration the point at which a buried pipe enters the soil either at grade or from a wall or discharge structure

Settlement the changes in volume of soil under constant load which result in the downward movement, over a period of time, of a structure or vessel resting on the soil

Virtual Anchor a point or region along the axis of a buried pipe where there is no relative motion at the pipe/soil interface

Nomenclature

a, b, c = quadratic equation functions

A = cross-sectional metal area of pipe, in^2

A_c = surface area of a 1-in long pipe segment, in^2

B_d = trench width at grade, in

C_D = soil bearing parameter from Table B4.15, dimensionless

C_k = horizontal stiffness factor for backfill [61],[2] dimensionless

D = pipe outside diameter, in

dL = length of pipe element, in

E = Young's modulus for pipe, psi

f = unit friction force along pipe, lb/in

f_{min}, f_{max} = minimum, maximum unit friction force on pipe, lb/in

F_f = total friction force along effective length, lb

F_{max} = maximum axial force in pipe, lb

H = pipe depth below grade, in

I = pipe section moment of inertia, in^4

k = soil modulus of subgrade reaction, psi

k_h = soil horizontal modulus of subgrade reaction, psi

$k_{i,j}$ = orthogonal soil springs on pipe, lb/in

k_v = soil vertical modulus of subgrade reaction, psi

L_1 = length of transverse pipe run, in

L_2 = length of longitudinal pipe run, in

L_m = minimum slippage length of pipe, in

L' = effective slippage length for short pipes, in

L'' = effective slippage length for long pipes, in

n = number of modeling elements for pipe springs, dimensionless

N_h = horizontal force factor,[61] dimensionless

P = maximum operating pressure in pipe, psi

P_c = confining pressure of backfill on pipe, psi

S_A = allowable expansion stress range, psi

SE = expansion stress, psi

S_h = basic material allowable stress at T degrees fahrenheit, psi

t = pipe wall thickness, in

T = maximum operating temperature, °F

T_o = ambient temperature of pipe, °F

w = soil density, pcf, pci

W_p = unit weight of pipe and contents, lb/in

α = coefficient of thermal expansion of pipe, in/in/°F

β = pipe/soil system characteristic,[62] in^{-1}

ε = pipe unit thermal expansion, in/in

μ = coefficient of friction, dimensionless

Ω = effective length parameter, in

1 inch = 25.4 mm

1 lb = 4.448 N

1 psi = 6894.8 Pa

deg F = 1.8 deg C + 32

Loads

Thermal Expansion. Thermal displacements at the elbows, branch connections, and flanges in a buried piping system and the forces and moments resulting from the displacements may be determined by analyzing each buried run of pipe by the method described in this section.

Installations with Continuous Runs. For buried piping installations that contain continuous runs without flexible couplings, the passive restraining effects of soil bearing on the transverse legs at the ends of long runs subject to thermal expansion may be significant and result in high axial forces and elbow or branch connection bending moments.

Installations with Flexible Couplings. For buried piping installations that incorporate flexible couplings into the pipe runs subject to thermal expansion, the bending moments and stresses may be substantially reduced. However, the flexible couplings must be chosen carefully to accommodate the thermal expansion in the pipe, and the friction forces or stiffness in the coupling must be considered.

Installations with Penetration Anchors. For buried piping systems in which the building penetration provides complete restraint to the pipe, it is necessary to calculate the penetration reactions to thermal expansion in the initial buried run. If this run incorporates flexible couplings, piping reactions at the penetration resulting from unbalanced forces due to internal pressure must be considered.

Installations with Flexible Penetrations. For buried piping systems in which the building penetrations permit some axial or angular movements, the interaction between the buried run outside the penetration and the point-supported portion of the system inside the building must be considered.

Pressure. Pressure loads in buried piping are important for two primary reasons:

1. In pipe runs which incorporate flexible couplings, there is no structural tie between the coupled ends, with the result that internal pressure loads must be reacted externally. External restraint may be provided by thrust blocks, external anchors, soil resistance to elbows or fittings at each end of the pipe run, or by control rods across the coupling. Where one or both of the ends terminate at a penetration or an anchor, or at connected equipment such as a pump or vessel, the pressure forces can be quite high and must be considered in the anchor or equipment design.

2. For discharge structures, the reaction forces due to upstream pressure and mass flow momentum in the discharge leg may be high and must be considered in the design of the last elbow or bend before the discharge.

Earthquake. An earthquake subjects buried piping to axial loads and bending moments from soil strain due to seismic waves, or from ground faulting across the axis of the pipe. The seismic soil strain can be estimated for a design earthquake in a specific geographical region, from which design values for forces and moments in buried piping can be calculated. However, consideration of the magnitude and effects of seismic ground faulting on buried piping is beyond the scope of this section.

Calculations

The calculations for stresses in restrained underground piping are carried out in four steps, as follows.

Assembling the Data. The pipe material and dimensions, soil characteristics, and operating conditions must be established:

Pipe Data

1. Pipe outside diameter D, in
2. Wall thickness t, in
3. Length of pipe runs L_1 (transverse) and L_2 (longitudinal), in
4. Young's modulus E, psi
5. Pipe depth below grade H, in

Soil Characteristics

1. Soil density w, pcf (from site tests)
2. Type of backfill
3. Pipe trench width at grade B_d, in
4. Range of coefficient of friction μ between pipe and backfill

Operating Conditions

1. Maximum operating pressure P, psi
2. Maximum pipe temperature T, °F
3. Ambient pipe temperature T_o, °F
4. Pipe coefficient of thermal expansion α, in/in/°F

Calculations of Intermediate Parameters. The following parameters must be calculated:

Maximum Relative Strain ε at the Pipe/Soil Interface, in/in. For thermal expansion, this is the unit thermal elongation of the unrestrained pipe,

$$\varepsilon = \alpha(T - T_o) \tag{B4.71}$$

where α = coefficient of thermal expansion
$T - T_o$ = difference between operating and installation temperatures

Modulus of Subgrade Reaction k, psi. This is a factor which defines the resistance of the soil or backfill to pipe movement due to the bearing pressure at the pipe/soil interface. Several methods for calculating k have been developed in recent years by Audibert and Nyman, Trautmann and O'Rourke, and others.[63–67] For example,[61] for pipe movement horizontally, the modulus of subgrade k_h may be found by

$$k_h = C_k N_h w D \text{ psi} \tag{B4.72}$$

where C_k = a dimensionless factor for estimating horizontal stiffness of compacted backfill. C_k may be estimated at 20 for loose soil, 30 for medium soil, and 80 for dense or compacted soil.
w = soil density, lb/in^3
D = pipe outside diameter, in
N_h = a dimensionless horizontal force factor from Fig. 8 of Ref. 61. For a typical value where the soil internal friction angle is 30°, the curve from Ref. 61 may be approximated by a straight line defined by

$$N_h = 0.285 H/D + 4.3$$

where H = the depth of pipe below grade at the pipe centerline, in

For pipe movement upward or downward, the procedures recommended in Ref. 63 may be applied. Conservatively, the resistance to upward movement may be considered the same as for horizontal movement with additional consideration for the weight of the soil. Resistance to downward movement may conservatively be considered as rigid for most expansion stress analysis.

Unit Friction Force at the Pipe/Soil Interface f.

$$f = \mu(P_c A_c + W_p) \text{ lb/in} \tag{B4.73}$$

where μ = coefficient of friction between pipe and soil
P_c = confining pressure of soil on pipe, psi
A_c = surface area of a pipe segment, in^2
W_p = unit weight of pipe and contents, lb/in

For piping which is buried within 3 pipe diameters of the surface, confining pressure P_c may be estimated by

$$P_c = wH \text{ lb/in}^2$$

where w = the soil density, lb/in
H = the depth below grade, in

For piping which is buried more than 3 pipe diameters below grade, confining pressure P_c is found by using the modified Marston equation[67]:

$$P_c = wC_D B_D \text{ lb/in}^2$$

where C_D = a dimensionless parameter obtained from Table B4.15
B_D = the trench width, with a maximum value of 24 in plus the pipe diameter

TABLE B4.15 Approximate Safe Working Values of C_D for Use in Modified Marston Formula

Ratio H/B_D	Damp topsoil and dry and wet sand	Saturated topsoil	Damp yellow clay	Saturated yellow clay
0.5	0.46	0.47	0.47	0.48
1.0	0.85	0.86	0.88	0.90
1.5	1.18	1.21	1.25	1.27
2.0	1.47	1.51	1.56	1.62
2.5	1.70	1.77	1.83	1.91
3.0	1.90	1.99	2.08	2.19
3.5	2.08	2.18	2.28	2.43
4.0	2.22	2.35	2.47	2.65
4.5	2.34	2.49	2.53	2.85
5.0	2.45	2.61	2.19	3.02
5.5	2.54	2.72	2.90	3.18
6.0	2.61	2.91	3.01	3.32
6.5	2.68	2.89	3.11	3.44
7.0	2.73	2.95	3.19	3.55
7.5	2.78	3.01	3.27	3.65
8.0	2.82	3.06	3.33	3.74
9.0	2.88	3.14	3.44	3.89
10.0	2.92	3.20	3.52	4.01
11.0	2.95	3.25	3.59	4.11
12.0	2.97	3.28	3.63	4.19
13.0	2.99	3.31	3.67	4.25
14.0	3.00	3.33	3.70	4.30
15.0	3.01	3.34	3.72	4.34
∞	3.03	3.38	3.79	4.50

Pipe/Soil System Characteristic[62]

$$\beta = [k/(4EI)]^{1/4} \text{ in}^{-1} \tag{B4.74}$$

where k = soil modulus of subgrade reaction k_b or k_v, psi
E = Young's modulus for pipe, psi
I = area moment of inertia for pipe, in^4

Minimum Slippage Length L_m[68]

$$L_m = \varepsilon AE/f \text{ in} \tag{B4.75}$$

where A = pipe cross-section area

Maximum Axial Force F_{max} in the Longitudinal Pipe Run. The maximum axial force in a pipe long enough for friction force to develop to the point where a region of the pipe is totally restrained longitudinally by the soil is found by

$$F_{\max} = f L_m = \varepsilon A E \text{ lb} \tag{B4.76}$$

Classification of the Pipe Runs

Purpose. The classification and subclassification of the buried pipe elements is used in choosing the proper equation for effective slippage length L' or L'' which is then used in calculating piping forces and stresses. The pipe segment identified by the dimension L' or L'' always begins at either an elbow, bend, tee, or branch connection and terminates at the point (described below as the *virtual anchor*) at which there is no slippage or relative movement at the pipe/soil interface.

Classification of the Pipe Elements. It is in the bends, elbows, and branch connections that the highest stresses are found in buried piping subject to thermal expansion of the pipe. These stresses are due to the soil forces that bear against the transverse run (the run running perpendicular or at some angle to the direction of the pipe expansion). The stresses are proportional to the amount of soil deformation at the elbow or branch connection.

Piping elements are divided into three major categories depending upon what type of transverse element is being analyzed, as follows:

Category A. Elbow or bend (see Fig. B4.27)

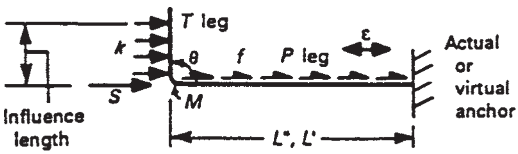


FIGURE B4.27 Element category A, elbow or bend.

Category B. Branch pipe joining the longitudinal run (see Fig. B4.28)

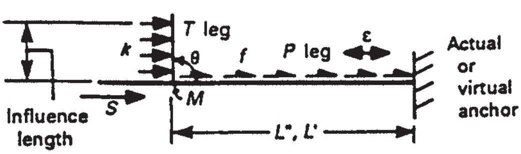


FIGURE B4.28 Element category B, branch pipe joining the P leg.

Category C. Longitudinal run ending in a tee (see Fig. B4.29)

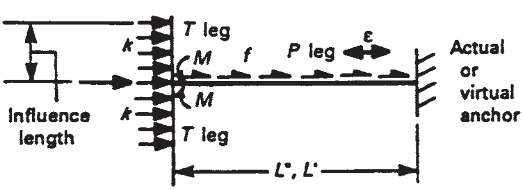


FIGURE B4.29 Element category C, tee on end of P leg.

Category D. Straight pipe, no branch or transverse run (see Fig. B4.30)

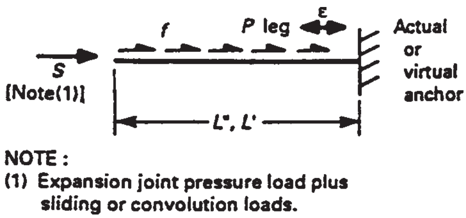


FIGURE B4.30 Element category D, straight pipe.

Categories A, B, and C are further divided into three subcategories depending on the configuration of the pipe run at the end opposite that being analyzed. The piping elements are classified as follows:

A1, B1, C1. Other end free or terminating in a flexible coupling or joint

A2, B2, C2. Other end contains an elbow or tee

A3, B3, C3. Other end is anchored

Category D elements include straight runs between an anchor (either actual or virtual) and a free end or a pipe section that is connected to an expansion joint.

The elements are further broken down into subtypes depending upon whether the longitudinal run (the pipe or P leg) and the transverse run (called the T leg) are long or short with respect to certain criteria. The transverse or T leg is the run against which the soil bears, producing an in-plane bending moment in elbow, branch, or tee. (Category D elements have no transverse leg.)

The strict criterion for a long or short transverse leg is whether the length of the transverse run L_1 is longer or shorter than $3\pi/4\beta$, the length at which the hyperbolic functions in Hetenyi's equations,[62] approach unity. The critical value for L_1 is often called the *influence* length, or that portion of transverse or T run which is deflected or "influenced" by seismic soil strain or pipe thermal expansion along the axis of the longitudinal or P run. In practice, a critical influence length L_1 of 25,983 to $1.2/\beta$ may often be used, since there is very little deformation or load in that portion of the transverse run which exceeds this length. This implies that the vast majority of the bearing load on the transverse or T leg occurs in the first several feet of the pipe at the bend or branch. In summary, a transverse pipe is "long" if

$$L_1 \geq 3\pi/4\beta \text{ (conservative)}$$

or

$$L_1 \geq 1/\beta \text{ to } 1.2/\beta \text{ (usually acceptable)}$$

The criterion for a short or long P leg is whether its length L_2 is sufficiently long to experience the maximum force that can develop at the friction interface. For full maximum friction force ($F_{\max} = \varepsilon AE$) to occur in a straight pipe axially free at each end, its length L_2 would have to equal or exceed $2L_m$ with L_m calculated by Eq. (B4.75). If one end terminates in an elbow or a tee, with the other end remaining axially unrestrained, the total length L_2 necessary for full friction to develop is $L'' + L_m$; the friction force over L_m is equal to the soil bearing force S plus the friction force acting on the length L' or L'', which is called the *effective slippage length*. The effective slippage length is the maximum length along which slippage occurs at the pipe/soil interface of a pipe with a transverse leg or branch. The effective slippage length L'' for long pipes with long transverse legs is calculated by

$$L'' = \Omega[(1 + 2F_{\max}/f\Omega)^{1/2} - 1] \text{ in} \tag{B4.77}$$

where $\Omega = AE\beta/k$ and $F_{\max}$ is calculated by Eq. (B4.76).

Equation (B4.77) applies to bends, tees, and branches. Although Eq. (B4.77) was developed for the case where $L_2 = L'' + L_m$, it applies also for any case where $L_2 > L'' + L_m$, since the length of the region where there is zero slippage at the friction interface is immaterial.[68] Using L'' as calculated by Eq. (B4.77), it can now be established that a P leg is classified long if it meets these criteria:

1. For Types A1, B1, C1, $L_2 \geq L_m + L''$;
2. For Types A2, B2, C2, $L_2 \geq 2\, L''$;
3. For Types A3, B3, C3, D, $L_2 \geq L''$.

That point which is located a distance L' or L'' from the bend, branch, or tee, is called the virtual anchor, since it acts as if it were a three-axis restraint on the pipe.

Locating the Virtual Anchor. Calculation of the forces and moments in buried piping at the changes in direction requires that the location of the virtual anchor (the effective slippage length L' away from the bend or branch element) in the P run and the deformation δ of the soil at the buried element be established. For elements of all types with long P legs, L'' may be calculated by Eq. (B4.77).

For Types A1, B1, and C1 elements (with one end of the P leg free or unrestrained axially) with "short" P legs, L' must be found by a less direct method as follows[68]:

$$L' = [-b + (b^2 - 4ac)^{1/2}]/2a \text{ in} \tag{B4.78}$$

where $a = 3f/(2AE)$
$b = \varepsilon - fL_2/(AE) + 2f\beta/k$
$c = -f\beta L_2/k$

However, the most highly stressed runs in a buried piping system typically are restrained at both ends, either by a combination of transverse runs or a transverse run and an anchor (either real or virtual).

For Types A2, B2, and C2 elements with short P legs, L' is expressed by

$$L' = L_2/2 \text{ in} \tag{B4.79}$$

For Types A3, B3, C3, and D elements with short P legs, L' is expressed by

$$L' = L_2 \text{ in} \tag{B4.80}$$

Computer Modeling of Buried Piping

Determination of Stresses. With f, k, and L' or L'' established, the stresses in a buried pipe due to thermal expansion can be determined with a general purpose pipe stress computer program. A buried piping system can be modeled with a typical mainframe or microcomputer pipe stress program by breaking the buried portions into elements of convenient length and then imposing a transverse spring at the center of each element to simulate the passive resistance of the soil. The entire pipe can be divided into spring-restrained elements in this manner; however, the only regions of the pipe that really need to be modeled in this manner are the lengths entering and leaving elbows or tees. The analyst should refer to the program users' manual for guidance in modeling soil springs.

All pipe stress computer programs with buried piping analysis options require that the following factors be calculated or estimated:

1. Location of the virtual anchor (dimension L' or L'')
2. Soil spring rate $k_{i,j}$, which is a function of the modulus of subgrade reaction k.
3. Influence length, also a function of k.

Some programs ignore the friction at the pipe/soil interface; this is conservative for calculating bending stresses on the buried elbows and branch connections, but may be unconservative for calculating anchor reactions.

Determination of Element Lengths. The element lengths and transverse soil spring rates for each element are calculated by the following procedure:

1. Establish the element length dL and the number n of elements, as follows:
(A) Set the element length to be equal to between 2 and 3 pipe diameters. For example, dL for a NPS 6 may be set at either 1 ft or 2 ft, whichever is more convenient for the analyst.
(B) Establish the number n of elements by:

$$n = (3\pi/4\beta)/dL \tag{B4.81}$$

This gives the number of elements, each being dL inches in length, to which springs are to be applied in the computer model. The number n of elements is always rounded up to an integer.

2. Calculate the lateral spring rate $k_{i,j}$ to be applied at the center of each element.

$$k_{i,j} = kdL \text{ lb/in} \tag{B4.82}$$

where k = the modulus of subgrade reaction calculated from Eq. (B4.72).

3. Calculate the equivalent axial load necessary to simulate friction resistance to expansion. The friction resistance at the pipe/soil interface can be simulated in the computer model by imposing a single force F_f in a direction opposite that of the thermal growth.

$$F_f = fL'/2 \text{ or } fL''/2 \text{ lb} \tag{B4.83}$$

4. Incorporate the springs and the friction force in the model. The mutually orthogonal springs $k_{i,j}$ are applied to the center of each element, perpendicular to the pipe axis. Shorter elements, with proportionally smaller values for the springs on these elements, may be necessary in order to model the soil restraint at elbows and bends. The friction force F_f for each expanding leg is imposed at or near the elbow tangent node, opposite to the direction of expansion.

Determination of Soil Parameters. Soil parameters are difficult to establish accurately due to variations in backfill materials and degree of compaction. Consequently, values for elemental spring constants on buried pipe runs can only be considered as rational approximations. Stiffer springs can result in higher elbow stresses and lower bending stresses at nearby anchors, while softer springs can have the opposite effects. Backfill is not elastic; testing has shown that soil is stiffest for very small pipe movements, but becomes less stiff as the pipe movements increase. References 61, 63, and 66 discuss soil stiffness and recommend procedures for estimating values for k which are consistent with the type of soil and the amount of pipe movement expected. The analyst should consult the project geotechnical engineer for assistance in resolving any uncertainties in establishing soil parameters, such as the modulus of subgrade reaction k, confining pressure p_c, and coefficient of friction μ.

Pipe with Expansion Joints. An expansion joint must be considered as a relatively free end in calculating stresses on buried elbows and loads on anchors. Since incorporation of expansion joints or flexible couplings introduces a structural discontinuity in the pipe, the effects of the unbalanced pressure load and the axial joint friction or stiffness must be superimposed on the thermal expansion effects in order to determine the maximum pipe stresses and anchor loads.

Pipe Stresses at Building Penetrations. Stresses at building penetrations can be calculated easily after the reactions due to thermal expansion in the buried piping have been determined. If the penetration is an anchor, then the stress due to the axial force F_{max} and the lateral bending moment M can be found by

$$S_E = F_{\max}/A + M/Z \text{ psi} \tag{B4.84}$$

If the penetration is not an anchor, but is instead a simple support with a flexible water seal, it is necessary to determine the stiffness affects of the water seal material in order to calculate the stress in the pipe at the penetration. Differential movement due to building or trench settlement can generate high theoretical stresses at piping penetrations to buildings. Calculation of such stresses is beyond the scope of this section.

Allowable Stress in Buried Pipe

Buried piping under axial stress can theoretically fail in one of two ways: either by column buckling (pipe pops out of the ground at midspan) or local failure by crippling or tensile failure (much more serious than column buckling). Since buried piping stresses are secondary in nature, and since the piping is continuously supported and restrained (see Fig. B4.31), higher total stresses may be permitted

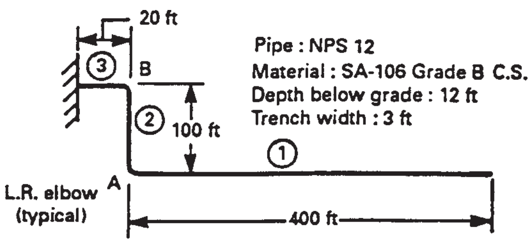


FIGURE B4.31 Plan of example buried pipe.

as follows:

$$S_C \leq S_A + S_h \tag{B4.85}$$

where S_A and S_h are as defined in Para. 102.3.2 of B31.1 Code.

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